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DESIGN CRITERIA FOR ROLLING ELEMENT AIR FRAME BEARINGS FOR HIGH TEMPERATURE AND HIGH ALTITUDE USE

TECHNICAL DOCUMENTARY REPORT ASD-TDR-62-900

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FLIGHT DYNAMICS LABORATORY
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NOTICES

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FOREWORD

The research work in this report was performed by Marlin-Rockwell Corporation, Jamestown, New York, for the Flight Dynamics Laboratory, Directorate of Aeromechanics, Deputy for Technology, Aeronautical Systems Division, Wright-Patterson Air Force Base, Ohio, under AF Contract No. AF33(616)-6650. This research is part of a continuing effort to obtain high altitude and high temperature airframe bearings for flight vehicles, which is part of the Air Force Systems Command's Applied Research Program 750A, the Mechanics of Flight. The Project Nr. is 1315, "Bearings and Mountings for Advanced Systems," and the Task Nr. is 131501, "Advanced Bearing Concepts." Lt. J. R. Clegg, Mr. P. C. Hanlon, and Capt. C. D. Stuber of the Flight Dynamics Laboratory were the Project Engineers. The research was conducted from July 1959 to July 1962 by Harold E. Munson, Jamshed B. Havewala, and John H. Johnson. The Contractors Report number is 1299.

Authors would like to acknowledge the invaluable assistance and guidance of Mr. W. G. Burgett, Mr. D. P. Johnston and Mr. A. S. Irwin during this investigation.

ABSTRACT

The investigation constitutes an experimental study in the field of aircraft control bearings directed toward evaluation of bearing roll configurations, bearing materials and bearing lubricants for operation in the temperature range of 1200°F at simulated altitude of 250,000 feet.

Four different roll designs, together with twelve different material combinations were investigated. Resulting best design and two best materials combinations (6B -vs- 6B and CA-3 -vs- 6B) were subjected to stresses up to 325,000 psi at the temperature and vacuum.

One inch diameter bore, self-aligning, double row roller bearings fabricated from 6B -vs- 6B carried loads to 5,000 lbs. (280,000 psi) for 40,000 cycles at 1200°F and vacuum. DF-700 dry film (MoS_2-Sodium silicate), when applied to all contact surfaces, gave superior performance among five different dry films investigated. Friction coefficients with the lubricants were in the range of .08 to .25. Successful bearing operation requires considerable deviation from design criteria for fluid lubricated bearings. This may have a very profound effect on control system design.

Test results suggest that a single row bearing of the BR-16 type could sustain the same loads with less torque. Further investigation of the single row design is desirable.

PUBLICATION REVIEW

This report has been reviewed and is approved.

FOR THE COMMANDER:

Ambrose B. Nutt

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INTRODUCTION

Recent advances in the field of missile and space travel have imposed requirements beyond the capabilities of currently used bearing materials and lubricants.

In previous experiments (1), ball bearings of the control type suffered severe decrease in life under high temperature-high load oscillatory conditions. The progressive failures encountered with ball bearings in this type of operation are accelerated by the fact that wear products produced during the initial oxidation of bearing surfaces cannot escape from the rolling-load area.

The substitution of roller bearings reduces the unit load between rolling element and raceways which in turn reduces the rate of friction oxidation. The concave roller type of bearing design, tends to promote purging of wear products from the contact areas.

With this design concept as the basis, research was undertaken to develop an aircraft control bearing for operation at 1200°F and a simulated altitude of 250,000 feet. The work involved the evaluation of roller design, determination of satisfactory bearing structural material, and evaluation of lubricants, as well as detail bearing design.

The enviromental factors of high temperature and altitude made evaluation of material and component design necessary. It was expected that altitude would have some adverse effect on the operation of rolling element airframe bearings. High temperature oxidation and friction oxidation of bearing parts may be reduced with the vacuum condition. However, the lack of an oxide film under a dry, running condition may create a problem of welding and tearing of rolling contact surfaces. This in turn tends to increase wear due to pick up of materials in the wear path. To promote escape of such wear products, various configurations of concave rollers were tested. Hot hardness, resistance to oxidation at high temperature, resistance to welding, and general compatibility with other materials were qualifications to be evaluated for bearing structural materials. The usefulness of dryfilms as lubricants was also evaluated.

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TEST SUMMARY

Because of the extensive nature of the testing, a logical sequence of evaluations was set up, each of which was a stepping stone to the succeeding one. These steps are referred to as "Phases" and may be briefly summarized as follows:

Phase I - Roller Design Criteria

This work involved an investigation of four different roller configurations. A simple hourglass design showed the best results and was selected for material evaluation.

Phase II - A-1 - Evaluation of Materials

Using the best roller design from Phase I, eleven different material combinations were studied, in addition to the combination used in Phase I. CA-3 vs. 6-B and 6-B vs. 6-B showed the best over-all performance as measured by condition of matching surfaces, low push-pull force requirements, low wear, and absence of apparent surface deterioration.

Phase II - A-2 - Materials Under Load

The two best combinations from Phase II - A-1 were tested under increasing radial load. These tests were run on single roll specimens. It was found feasible to use these materials at loads corresponding to bearings rated at 50 per cent of 5000 NDL for limited life requirements. (This is approximately 1000 lbs. load on a roll, 260,000 psi).

Phase II - B - Coatings and Treatments

Four dry films were evaluated using 6-B rolls coated with the test material running against CA-3 and 6-B cylinders. CaF_2-Oxide Frit coating was one which prevented metal to metal contact.

Later, during testing of complete bearings, DF-700 dry film was found to have promise under high vacuum and was tried on four bearings. It appeared more satisfactory than did the CaF_2.

Phase III - Testing of Bearings

The results from this phase indicate that limited performance can be accomplished with the BR-16 type double row configuration, if the following design criteria are used:

1. Roll race curvature should be 150% of race curvature.

2. Number of rolls should be reduced from the original 26 to provide extra stock in the cage.

3. Outer diameter of the cage should be tapered to avoid interference with the face corner of the outer race.

4. Bearing should have an internal clearance of .004".

5. All parts should be treated with DF-700 dry film coating.

6. Loads should be less than 2000 NDL (i.e., about 5000 lbs. radial load on this modified bearing; this is equivalent to 280,000 psi)

Although this bearing has carried 5000 lbs. for a short time, results suggest that a single row bearing of BR-16 type could sustain the same load level with less friction. This type of single row bearing will be capable of withstanding thrust of the magnitude of 10% of the radial load. Further investigation of single row bearings appears desirable.

TEST EQUIPMENT

Figures 2 and 3 show the test rigs for evaluation of specimens. The experimental facility is shown in Figure 4.

The equipment consisted of:

1. A vacuum chamber 4 feet in diameter, 5 feet long, with eight view windows, two push-pull feed-throughs, one rotary feed-through, thermocouple and electrical feed-throughs.

2. An NRC 100 cu. ft. per minute two stage rotary gas ballast vacuum pump.

3. An NRC Alphatron ionization gage.

4. A Minneapolis-Honeywell recorder and controller with associated valves for vacuum chamber pressure regulation.

5. A 12 kw furnace installed in the vacuum chamber surrounding test specimens.

6. A 1 hp U. S. Varidrive motor for oscillating the rolling test specimens.

7. A Baldwin Type U-1 2000 lbs. capacity load cell for measuring the actuating force during oscillation.

8. A Model 127 Sanborn Strain Gage Recorder for use with the load cell.

9. A Leeds and Northrup Micromax Recorder for temperature measurement.

10. A nut cracker type loading mechanism in the vacuum chamber with linkage through the two push pull seals for (1) oscillation of test specimens and (2) external variation of radial load.

The nut cracker loading fixture is equipped with a cam driven rod moving a central test cylinder back and forth between two test rolls which were loaded by a fixed bottom cylinder and an upper cylinder mounted in the movable "jaw". It was found necessary to provide guide rolls for the central cylinder and guide plates for the test rolls to keep the load application in a vertical line. Rolls tended to skew without this support.

For testing of bearings the cam driven rod moved a central plate back and forth between the two bearings, causing them to oscillate.

TEST PROCEDURE

Prior to each test, rolls were checked dimensionally and weighed on an analytical balance.

After assembly the chamber was evacuated and the test area was heated. It was found necessary to heat the system gradually to maintain the desired low pressure, which otherwise would have been affected by too rapid out gassing of the system under heat.

After obtaining a temperature of 1200°F and a simulated altitude of 250,000 feet, the drive motor was turned on to oscillate the specimens. At the same time the load was applied.

Actuating forces and wear measurements (the latter from movement of the external load mechanism) were recorded at frequent intervals. Tests of rolls and cylinders, Phases I and II, were run for 72,000 cycles or apparent failure. Tests of bearings were discontinued after 40,000 cycles.

Following the completion of oscillation, the test rig was left under vacuum until temperature had dropped to 200°F. After disassembly, test specimens were re-weighed and measured dimensionally.

DISCUSSION

Phase I - Roller Design Criteria

Four different roller configurations "A", "B", "C", and "D" in Figure 1 were tested. The best design from this phase was used in the testing for material evaluation and in the final development of the BR-16 internally self-aligning roller bearing.

Roll Design "A"

This is a standard hourglass design in which the curved contact surface has been extended in such a fashion that under limited applied load the contact ellipse falls within the curvature.

Roll Design "B"

This configuration provides a gap at the center of the roll and has the same possible total contact area as "C". In this case, the wear products may be removed from the roll paths on both sides and in the center.

Roll Design "C"

This design is similar to "A". However, curvature contact surface is reduced to achieve 20 to 50% overlap contact ellipse area under the applied load. In this configuration, the entire width of the concave surface of the roll makes contact with the race.

Roll Design "D"

This grooved type of design provides four gaps and five contact surfaces with a total available contact area about the same as in "B" and "C". This configuration provides a maximum possible allowance for the removal of wear products.

All rolls in this phase were made of Star-J and cylinders were of M-252. Eighteen tests were run on the four roll configurations with the results shown in Table No. 1.

All tests, except No. 9, were complete runs of 72,000 cycles (24 hours). These results are plotted in Figures 7 through 14. Test No. 9 was terminated prematurely due to vibration in the rig.

In most of these runs it was observed that there was a sharp variation in the friction (push-pull force) as well as a rapid increase in total wear during the early stage of the test runs. Such early variation was probably caused by the rolls tending to adjust themselves to the most suitable position. After a comparatively small number of cycles the running tended to "smooth out", which is noticeable in most of the test runs.

In some tests consistent data were not obtained on strain due to effect of heat, misalignment, and gradual wearing out of the push-pull seals. Test Nos. 6 and 16 showed relatively high values of strain, due mainly to rubbing of the oscillating rod on the seal.

After a smoothing out period, all tests showed relatively the same rate of wear. Test Nos. 5 and 13 showed variation in the wear data (Figures 8 and 9). During these periods, malfunctions of the heating furnace caused variations in temperature, resulting in expansion or contraction of the leverage system and consequent erratic wear measurements.

Change in Roller Weight

In the majority of cases, rolls showed a decrease in weight. Roll design "D" showed the greatest loss which could be due to its four groove configuration allowing wear particles to escape. "C" rolls showed very little change in their weight. "C" rolls in Test No. 14, using longer stroke, showed a positive increment in weight possibly due to building up of wear or metal pick up from the guiding separators.

Appearance

In earlier tests a dark green film was noticed on specimens. Such coloration may have been caused by absorption of vapor particles from grease purging into the vacuum chamber through the push pull seals. New seals were installed on the push pull rod and no color change was observed in subsequent tests.

When observed under microscope, specimens showed crude or rough surfaces due to spalling or "metal picking" with the exception of Test Nos. 11, 12, and 13 on "C" rolls. These rolls, as well as matching cylinders, showed very smooth surfaces compared with any other test specimens. Test No. 14 showed a dark film on the test surfaces. However, appearance of these surfaces was very smooth except at some skidding spots.

"D" rolls showed very severe pits in the cylinder surfaces and have shown the worst wear surface indications among all four. The digging of cylinder metal was as much as .004" deep. Although these rolls gave maximum allowance to eliminate wear, they gave poor results on cylinder surfaces, possibly due to high corner stresses.

The cylinder surface in the evaluation of "B" rolls gave generally good results. However, in Test No. 5 pits about .0035" deep occurred. Those pits were inspected by a Pratt & Whitney comparator.

Figures 5 and 6 are the photographs showing the wear surfaces of the rolls and their matching cylinders after investigation.

In the conclusion of this phase the results are:

1. Roll design "C" gave best results showing smooth wear surfaces as well as a negligible change in the roll weights. When degree of oscillation was increased on these rolls, to provide overlapping of paths on the rolls, results were also good. This design was selected for material evaluation.

2. Roll design "A" produced rough surfaces with a wear build-up on the cylinder surfaces.

3. Roll design "B" gave generally good results. However, in one test surface pits developed.

4. Roll design "D" gave very poor results on the cylinder wear surfaces with pits as deep as .004". These rolls also suffered the greatest loss in weight.

Phase II - A-1 - Evaluation of Material

Several types of materials were considered as candidates for bearing structural components in the high temperature-high altitude field. These are listed in Table II, grouped according to their chemical and physical properties. On the basis of M-R-C Research Laboratory experience and a literature survey, eight materials and twelve combinations of these materials were selected.

These materials are:

Star-J	-	Cobalt base (cast)
6-B	-	Cobalt base (wrought)
M-252	-	Nickel base (used only in Phase I)
Rene' 41	-	Nickel base (wrought)
CA-3	-	Tungsten carbide
K-162-B	-	Titanium carbide
608	-	Chromium carbide
AD-99	-	High density aliumina

These materials were run in the following combinations:

Rolls		Cylinders	
Star-J	vs	M-252	(Phase I only)

Rolls		Cylinders
AD-99	vs	K-162-B
6-B	vs	Rene' 41
CA-3	vs	6-B
6-B	vs	AD-99
Star-J	vs	608
Rene' 41	vs	CA-3
AD-99	vs	CA-3
6-B	vs	6-B
Rene' 41	vs	Rene' 41
K-162-B	vs	K-162-B
AD-99	vs	AD-99

In Phase II - A-1, twenty four tests of 72,000 cycles (24 hours) duration, were run evaluating the latter eleven combinations. Table III summarizes the results of this testing. Results on wear and friction of these tests are plotted in Figures 15 through 28.

CA-3 rolls versus 6-B cylinders and 6-B rolls versus 6-B cylinders showed better results and wear surfaces than any of the other combinations. AD-99 versus CA-3, 6-B versus AD-99, and AD-99 versus AD-99 also showed good wear surfaces but either friction forces, wear levels or corner stresses were higher for these three combinations.

The several other combinations suffered an appreciable degree of surface deterioration.

Instrumented data of Test No. 13 (CA-3 versus 6-B) were affected adversely by a looseness of the test assembly which developed during the run. Surfaces of the specimens in this test were in very good condition despite the added motion.

Friction forces and wear readings for all tests were lower than project requirements.

All specimens showed slight discoloration, presumably due to a thin oxide film formation. There is sufficient oxygen at a simulated altitude of 250,000 feet (2×10^{-2} mm Hg pressure) to permit oxidation. CA-3 was the most sensitive to oxidation of the materials tested.

It was to be noted that the material combinations of Star-J versus M-252, Star-J versus 608, AD-99 versus K-162-B and K-162-B versus K-162-B, which have shown good results (3) (4) at high temperature and atmospheric pressure, showed a high degree of surface deterioration in vacuum.

On the basis of these results, 6-B versus 6-B and 6-B versus CA-3 were chosen as bearing material combinations and subjected to further evaluation.

Phase II - A-2 - Materials Under Load

Fifteen (15) tests were run on CA-3 (rolls) versus 6-B (cylinders), 6-B versus 6-B, and 6-B versus CA-3, results of which are shown in Table IV. Number of cycles versus wear and friction during these tests are shown on Figures 29 through 34. Load versus average friction as shown on Figure 35.

Test No. 1 (CA-3 rolls versus 6-B cylinders) was terminated after 1200 cycles due to the generation of a high degree of vibration in the system. Rolls showed severe metal pick-up and extensive digging into the cylinder surfaces. Coefficient of friction was higher than Phase 2. Test load was 2000 lbs. (325,000 psi).

Test No. 2 which was also run with the same material combination, but at 1000 lbs. load (260,000 psi) showed similar but less serious effects. The test was terminated at 4000 cycles due to excessive wear.

Tests No. 3 and 6, on the same material combination, but at 500 lbs. (200,000 psi) load, were complete runs of 72,000 cycles. Though friction forces and rate of wear were moderate, some cylinder surface digging by the corner of the rolls was visible. This is apparently due to CA-3 being a very hard material as compared with 6-B.

Test No. 12, also on the same material combination, but using the "A" type roll design to reduce average Hertz stresses, was run under 1000 lbs. load. Early failure was encountered due to misalignment in the rig.

Digging of the roll corners into the cylinder surface occurs under high loads when the contact ellipse overlaps the available roll surface area. (See Figure 36). Usual practice in this type of bearing design is to provide a roll surface area which will result in 50% ellipse overlap at maximum loads. However, this design approach is not always acceptable in the case of high temperature materials under present test conditions when the roll material is very hard compared with the corresponding cylinder (race) material.

If larger roll surface area is provided such that the contact ellipse falls within this available surface area under a particular applied load, such effects as described above could be eliminated. However, under dry running conditions, the wear debris will tend to build up between the roll and race surfaces outside the contact area, which in turn will create high abrasive rubbing and extensive corner loading.(See Figure 36). This situation leads to a possible conclusion that for each particular load a single bearing internal geometry is necessary to avoid the above described corner effects. That is, a bearing should be used for only one particular load.

However, testing indicates that this problem may be relieved by proper material selection. As previously mentioned, relative hardness of the materials plays an important role in corner stress effects. When 6-B versus 6-B combination was tested in Tests No. 4 and 8, under 500 lbs. load, no excessive surface damage was noted.

Successful results were also obtained in Tests No. 5, 7, 9, and 10 when 6-B versus CA-3 combination was used instead of original CA-3 (rolls) versus 6-B (cylinders). Because 6-B is a softer material, the corner stress effects were considerably reduced. All tests on these two combinations were complete runs of 72,000 cycles except Test No. 9, which was terminated at 45,000 cycles. During this test, the nut locking the push pull rod loosened, allowing the cylinder to rotate. This shifted the rolls and resulted in the ultimate failure of the test. Both rolls were badly damaged and highly rubbed on the sides by the spacer. This resulted in high weight loss.

Test No. 7 at 57,000 cycles showed a sudden jump in wear which could have been caused by the change in position of the wear indicator due to disturbance by some external means. No other reason for this sudden change could be found.

Difficulty in the heating unit was encountered in Test No. 10 at about 60,000 cycles. Seven heaters were found to be burned out. By the end of the test, temperature had dropped to 1050°F. This drop in temperature causing rig contraction and, therefore, realignment of contact surfaces could be the reason for sudden increase in friction. Both roll and cylinder show good surfaces.

Test No. 10 shows rapid wear up to 40,000 cycles after which it levels off. This may be due to corners of the roll gradually relieving or wearing away until stress concentrations were accommodated.

Test No. 11 and 13 were run using 6-B versus 6-B combination under 1000 lbs. load. Test No. 11 was terminated at 1400 cycles due to bending of the push pull rod. This bending may have been caused by possible misalignment in the system. Roll or cylinder surfaces show no apparent damage. Test No. 13 was a successful run of 72,000 cycles. During the early stage of this test, some fluctuation is noted in the wear data. Shifting of the rolls probably caused this scatter.

"A" Roll design was used in Tests No. 14 and 15 to study higher loads. Test No. 14 (6-B versus 6-B) was run through 72,000 cycles under 1500 lbs. (300,000 psi) load. Though friction force and wear rate were not high, the matching surfaces showed high degree of metal picking. Test No. 15 (6-B versus CA-3) was terminated at 7900 cycles. High degree of wear was encountered in this run.

It is concluded from the work done in this phase, that:

At 500 lbs. load all tests showed moderate friction and rate of wear. Under increasing loads 6-B versus 6-B showed better matching surfaces than 6-B versus CA-3.

Use of 6-B as roll material instead of CA-3 has relieved high corner stress concentration under increasing loads.

It is feasible to consider the use of these materials in bearings rated at 50% of 5000 NDL, for limited life requirements. (Approximately 1000 lbs. load on roll, (260,000 psi).

Phase II - B - Coatings and Treatments

Seventeen (17) tests were run, on CaF_2 (2), dry film 811, dry film 100-1, and CDL-5940 coatings. Test No. 16 was discarded because of difficulties encountered in the test rig. All tests are complete runs of 72,000 cycles under 500 lbs. load (200,000 psi). Results are shown in Tables V and VI and Figures 37 through 44.

These coatings prevented any high degree of metal to metal contact and thus reduced the effects of metal picking and welding. CaF_2 Oxide Frit dry film provided the best separating film of all. However, applied thickness and weight of this coating was considerably greater than the others. A large amount of dull green powder accumulated during the testing of this coating. This coating also showed very rough surface ranging from 120 to 170 rms micro-inches before running. However, it has been found that with proper technique of application and treatment, this problem can be eliminated. This is very important in holding proper internal geometries of the BR-16 type bearing.

CDL-5940 was the second best under these conditions. This coating did show slight pitting or welding. However, it appears that for extended life this coating may work better than CaF_2. Use of this coating in full-scale bearings in addition to CaF_2 was considered, but the very high cost of treating the sample bearings prevented its consideration for further investigation at this time.

Both dry film 100-1 and 811 prevented metal to metal contact to a certain extent, but surfaces were not as good as those observed with CaF_2 coating.

CA-3 cylinders (races) in all the tests showed very fine surfaces with slight material (coating) build up at various spots.

Wear data of Tests No. 3, 6, 11, and 17 show shifting of the readings during the early period. These changes are thought to be either due to rolls self-adjusting for appropriate position or due to breaking of dry film or due to some external disturbance affecting the indicator reading. In Test No. 17, at approximately 45,000 cycles, temperature started dropping when five heaters burned out. By the end of the test, temperature had dropped 110°F.

Friction data of test No. 15 are not plotted because one of the tubes in the recorder was found to be malfunctioning.

Thickness of the coating, surface finish and weight are listed in Table VI. Dry film 811 and 100-1 coated rolls show a decrease in weight after coating. It is believed that such loss occurred during the sand blasting process, prior to coating.

Following completion of this phase of testing, dry film DF-700 (5) was found to have promise as a lubricant in ultra-high vacuum. This lubricant then received consideration for Phase III.

Phase III - Testing of Bearings

The following combinations were selected for test bearings:

Bearing designation	BR-16-5	BR-16-6
Inner and outer rings	6-B	CA-3
Rolls	6-B	6-B
Retainer	Ni-Resist	Ni-Resist

All tests were conducted on the BR-16-5 bearings.

A static load test was conducted on a BR-16-5 bearing to determine radial load capacity at 1200°F. It was established that visible brinelling occurred above 4000 lbs. (225,000 psi) with heavy brinelling at 6000 lbs. (250,000 psi). Precise determination of the brinell point was impossible due to progressive distortion of the load mechanism under temperature and load, as well as discoloration of surface due to oxidation.

For dynamic testing of these bearings the following procedure was established:

Prior to assembly all critical dimensions of the bearings were checked and recorded. Parts were then ultrasonically cleaned.

When dry film lubrication was used, cages were sand blasted prior to coating. In order to obtain good adherence of the CaF_2 film to the cobalt base parts, 2% to 5% of sodium silicate was added to the dry film slurry.

After cleaning or dry filming, the bearings were assembled and installed in the test rig. Bearings were then heated gradually under vacuum and load was applied after environmental test conditions were attained, as described in Test Procedure.

A total of 21 tests were conducted on BR-16-5 bearings in the development of a satisfactory self-aligning aircraft control bearing. Several variations were made in retainer design and roll geometry. Figure 45 shows the basic BR-16 bearing design. Figure 46 shows the proposed single row BR-16 bearing which is recommended for further evaluation. Figure 47 shows the design of roll types used in the course of this investigation.

Table VII provides a summary of dynamic tests carried out in this phase. Figures 48 through 53 demonstrate bearing wear and friction versus test duration for each of the tests. It must be observed that radial loads varied from 45 lbs. in some tests to 5000 lbs. in Tests 17, 18 and 21.

At the completion of testing, A BR-16 bearing had been developed which was proven capable of limited performance under high temperature and altitude environmental conditions.

Problems encountered during evaluation of the BR-16 design and steps taken to correct those difficulties are as follows:

Test No. 1 - Bearings were run with no lubrication for 40,000 cycles under a radial load of 2000 lbs. (175,000 psi). The bearings were locked up at the end of the test. Inspection showed severe internal loading with contact area around the entire circumferences of the races. This contrasts with a normal radially loaded bearing which shows contact area through about 1/5 of the outer race circumference.

This internal loading was caused by the tendency of the unloaded rolls to move axially outward until internal clearance no longer existed. With rotation the rolls which shifted outward tend to roll in a new plane and to skew toward the outside even more. The rolls which were initially loaded in the normal position also tended to spread apart as rotation imparted a skewing moment to the rolls. The result was a wedging action within the bearings. From the appearance of the bearing surfaces, stresses due to internal loading exceeded the comparative strength of the material.

In an attempt to alleviate the condition, side plates were added to the bearing cages.

Test No. 2 - This test was a check of theefficiency of cage side plates under the same conditions as Test No. 1. Skidding and locking were not cured but friction and wear were reduced.

To improve performance further, outer races were opened to increase internal clearance by .0015.

Test No. 3 - This test was a check of the change in radial clearance. The test was discontinued when it was evident that no improvement had resulted.

The .0015 change in internal clearance was discarded. It was decided to determine the effect of a dry lubricant on bearing performance.

Test No. 4 - This test used CaF_2 Oxide Frit dry film on rolls and cages. Bearing friction values were somewl it improved but bearings were locked at the end of the test.

It was decided to run a set of BR-16 bearings at minimum radial load and room temperature to identify the cause of internal bearing distress.

Test No. 5 - These bearings were oscillated with no lubrication at 45 lbs. radial load and room temperature - atmospheric pressure to determine if the unsatisfactory operation experienced in the earlier tests was a function of bearing design, loading, or environment. Friction values rapidly increased, indicating that the bearings were following the characteristic pattern noted earlier. The test was discontinued after 1600 cycles to prevent damage to the bearings.

Analysis of results indicated that if the coefficient of friction between rolls and races could be reduced sufficiently, the rolls would slip axially inward against the skewing action and prevent internal loading. This could be checked by using oil lubrication.

Test No. 6 - The same bearings used in Test No. 5 were run again under minimum load and room temperature - atmospheric pressure environment, but with oil lubrication. Bearing performance was satisfactory.

It was decided to increase the load to 2000 lbs. but to maintain other conditions the same as Test No. 6.

Test No. 7 - This test with oil lubrication and a 2000 lbs. load was successful. Friction values remained normal and wear was not measurable.

Since dry film lubrication does not reduce friction to the degree that oil lubrication will, it was decided to approach the problem from the standpoint of bearing geometry. At the suggestion of ASD, two bearings were made up using rolls with the radiused section reduced from 0.375" wide to 0.190" wide. (See Figure 47).

Test No. 8 - Bearings were run dry at 45 lbs. load and room temperature, atmospheric pressure. After bearing performance had stabilized, load was increased to 500 lbs. Friction was moderate and the test was discontinued to permit evaluation of the roll design at high temperature and vacuum.

Test No. 9 - Bearings from Test 8 were run without lubricant at high temperature and simulated high altitude. When radial load was increased to 500 lbs. load, friction rapidly became excessive. When the test fixture was disassembled the bearings were locked.

It was decided to use rolls with a curvature of 1.064". (See Figure 46 and 47).

Test No. 10 - The bearings with increased curvature rolls were run dry at 45 lbs. load and room temperature, atmospheric pressure. Bearings operated satisfactorily.

Test No. 11 - The bearings from Test 10 were run with lubrication at high temperature and vacuum under 500 lbs. radial load. The test completed 40,000 cycles without failure. Bearings were tight after running but could be rotated by hand after tapping.

It was decided to use the same bearings for another test at high load to further evaluate this design.

Test No. 12 - The bearings from Test No. 11 were installed in the rig after being reoriented to provide unused surfaces in the loading zone. The test was run without lubrication at high temperature and vacuum under 2000 lbs. radial load. The test was discontinued due to excessive friction. The bearings were locked when the rig was disassembled.

It was decided to increase the radius of curvature of rolls to 1.497".

Test No. 13 - Bearings incorporating 1.497" radius rolls were run without lubrication at room temperature, atmospheric pressure and 45 lbs. radial load. Performance was satisfactory.

Test No. 14 - The bearings from Test No. 13 were run without lubrication at high temperature and vacuum. After successful operation at 500 lbs., the load was increased to 2000 lbs. Friction rose rapidly and the test was suspended. One of the bearings was found to have a broken cage with two "fingers" crushed into the roll path. The other bearing was in good condition but showed marks of skewing.

It was evident that bearing performance had been improved by the incorporation of increased radius of curvature in the rolls. However, large forces were still imposed on the cage by the skewing moment. New cages were designed to provide for eight rolls per row instead of the original thirteen. The length of the rolls was reduced from .5354" to .415" to permit a heavier center section of the cage. The reduced length of rolls did not affect the load carrying capacity.

Test No. 15 - Bearings with newly designed cages were run without lubrication at room temperature, atmospheric pressure under 45 lbs. load. Performance was satisfactory.

Test No. 16 - Bearings from Test No. 15 were run without lubrication at high temperature and altitude. After satisfactory performance at 500 lbs., the load was increased to 2000 lbs. The test completed 40,000 cycles after which the bearings were rough but free to rotate. Races and rolls showed torn surfaces. High rubbing and metal picking was visible in the loaded zone of the cage pockets but the condition of bearings was better than in any previous 2000 lbs. load testing.

Since the bearings were still capable of further operation, they were reassembled in the rig in such a manner as to provide fresh contact surfaces.

Test No. 17 - The bearings from Test No. 16 were run without lubrication at high temperature and altitude. After 3000 cycles at lesser loads, 5000 lbs. was applied to the bearings. The test was discontinued after the cam rod in the driving mechanism had broken twice due to high friction torque. Bearings were free to oscillate by hand and contact surfaces showed similar but slightly heavier contact stresses to those observed in Test No. 16. The O.D. of one of the cages showed rubbing with the face corner of the outer race.

To eliminate one of the problems observed in Test No. 17, the corners of the O.D. of the cages were tapered to prevent contact with the outer rings.

Test No. 18 - Bearings were run without lubrication at high temperature and vacuum. The bearings operated at 5000 lbs. load but with excessive vibration and friction resulting in premature suspension of the test. Both bearings exhibited metal pick-up and general distress.

Structural bearing changes permitted marginal operation of the BR-16 bearing. However, performance was not satisfactory at 5000 lbs. load and it was apparent that lubrication was necessary for improvement.

Test No. 19 - Rolls for the two bearings were treated with CaF_2 Oxide Frit. Because of difficulties encountered in coating the Ni-Resist cages, these were not treated. The bearings were run at high temperature and vacuum for the planned duration, 40,000 cycles, under 2000 lbs. load. Testing exhibited somewhat higher friction than some previous unlubricated tests at this load. Rolls were very badly worn due to severe rubbing in the cage pockets.

Because of the indifferent success with CaF_2 Oxide Frit and because of the promise of DF-700 (MoS_2 + graphite + silicate), it was decided to try the latter dry film.

Test No. 20 - DF-700 was applied to rolls and cages. Bearings were run at high temperature and vacuum under 2000 lbs. load. Friction increased gradually, probably due to eroding of the dry film. The condition of the bearings after the test was far better than any others tested at this environment and load. Rolls and races in the loaded zone showed metal to metal contact.

Because of the success of this test it was decided to conduct a high load test with DF-700 on all bearing surfaces.

Test No. 21 - DF-700 was applied to rolls, cages and races. Bearings were run at high temperature and vacuum. After 7,000 cycles at lesser loads, 5000 lbs. was applied to the bearings. The test completed the planned 40,000 cycles. During the last 2,000 cycles there was a sharp rise in friction and heavy wear occurred. Corners of the loaded zone rolls were slightly turned over due to loading. Cages were in excellent condition and bearings were free to rotate by hand. The stress exceeded the compressive strength of the material.

CONCLUSIONS AND RECOMMENDATIONS

The research and development described in this report has produced an aircraft control bearing suitable for operation at 1200°F and an altitude of 250,000 feet. In addition, valuable information has been gained concerning compatibility of materials in vacuum and effectiveness of various dry film lubricants. The following design criteria for the double row BR-16 type bearing have established:

1. Roll race curvature should be 150% of race curvature.

2. The number of rolls should be reduced instead of the original 26 to provide additional stock in the cage.

3. Outer diameter of the cage should be tapered to avoid interference with the face corner of the outer ring.

4. Bearing should have an internal clearance of .004".

5. 6-B is a satisfactory material for rolls and rings, and Ni-Resist is a satisfactory cage material for these conditions.

6. All bearing parts should be treated with DF-700 dry film coating.

7. Loads should be less than 2000 NDL (280,000 psi), (i.e., about 5000 lbs. radial load on this bearing).

This design is worthy of further development and testing. The BR-16 design incorporating CA-3 rings and 6-B rolls should be tested, using the specifications for roll and cage geometry just described.

In addition, evaluation of the single row concave roll bearing design, BR-16-7, should be made. Analysis of results of double row bearing testing indicates that the single row design should sustain the same load level with less friction. In addition, the single row bearing should be capable of withstanding an axial load equal to 10% of the radial load. This design is particularly worthy of development since it is a simpler and less expensive approach to the problem of an aircraft control bearing for high temperature and altitude applications.

LIST OF REFERENCES

1. "High Temperature Airframe Bearing Research Program" May 21, 1957; MRC Research Lab. Jamestown, New York

2. "Lubrication Properties of Some Bonded Fluoride and Oxide Coatings for Temperature to 1500°F" by Harold E. Slixley, July 28, 1960. NASA TN D478

3. "High Temperature Dry Bearing Research" by J. B. Havewala May 1960, MRC Research Lab. Jamestown, New York.

4. "Development of a Ceramic Roller Bearing" by K. M. Taylor and F. A. Savlino; February, 1961; WADD TR 60-684

5. "Lubricating Properties of Metal-Free Phthalocyanine and Metal Free Porphyrin in Simulated Space Enviroment" by E. C. Jackson, July 11, 1962 NRC Cambridge, Mass.

APPENDIX

TABLE I
PHASE I
TEST RESULTS

Test No.	Roll Design	Coefficient of Friction f Initial	Max.	Final	Wear In.	Change in + Weights gms.	Remarks
1	A	.060	.062	.030	.0046	+.0011, +.0017	Rough sur-
2	A	.030	.080	.060	.0040	-.0021, -.0075	faces. Build
3	A	.060	.120	.060	.0045	-.0043	up on
4	A	.030	.038	.023	.0021	-.0009, +.0002	cylinders
5	B	.015	.068	.030	.0080	-.0025, -.0033	Fair, but
6	B	.105	.150	.090	.0035	-.0027, -.0033	showed pitting
7	B	.068	.075	.045	.0016	-.0009, -.0011	on cylinders
8	B	.045	.075	.045	.0053	-.0005, -.0003	
9*	C	.090	.225	.085	.0081	+.0008, +.0013	Smooth
10	C	.090	.090	.015	.0083	+.0010, +.0022	surfaces,
11	C	.038	.075	.016	.0032	-.0002, +.0004	Negligible
12	C	.045	.075	.060	.0025	-.0001, +.0004	build up
13	C	.015	.030	.023	.0023	+.0002, +.0004	
14**	C	.240	.240	.060	.0012	+.0020, +.0021	
15	D	.053	.175	.042	.0090	-.0097, -.0076	Deep pits
16	D	.128	.150	.105	.0013	-.0058, -.0102	in the
17	D	.090	.090	.045	.0033	-.0126, -.0017	cylinders
18	D	.120	.128	.060	.0060	-.0028, -.0024	

Star J Rolls, M-252 Cylinders

72,000 cycles, 250,000 ft, 1200°F, 200 Lbs. Load (150,000 psi)

50 oscillation/min.(.500" stroke)

+First figure refers to top roll, second to the bottom

*10,102 cycles only

**2.18" stroke

TABLE II, VARIOUS MATERIALS AND THEIR COMPOSITION

Material	Chemical Composition %	Hardness at °F	Density lbs/in^3	Ultimate Tensile Strength	Compression Strength psi	Transverse Rupture Strength psi	Coefficient of Thermal Expansion $x\ 10^{-6}$ °F
Group I							
CA-3	WC - 94 Co - 6.0	Ra 91	.530			250,000	
#779	WC - 91 Co - 9.0	Ra 89.3 1200-80	.529		600,000	300,000	78° -1500 3.23
K-96	WC - 94 Co - 6	Ra 92	.535		800,000	250,000	-1200 2.5
Group II							
CA-815	Cr_3C_2-85 Ni - 15	Ra 88 1200-82.5	.253		420,000	110,000	68-1500 6.59
#608	Cr_3C_2-83 WC - 2.0 Ni - 15	Ra 88.3 1200-84	.253		500,000	113,000	76-1500 6.30
CR-83	Cr_3C_2-83 WC - 2 Ni - 15	Rc 72	.253		420,000	100,000	-1200 5.0
Group III							
6-B	Cr - 30 W - 4.5 Ni - 3.0 Fe - 3.0 Co - Bal C - 1.1	38-40 Rc	.303	165,000	240,000		32-1292 8.8

Material	Chemical Composition %		Hardness at °F	Density lbs/in^3	Ultimate Tensile Strength	Compression Strength psi	Coefficient of Thermal Expansion x 10^{-6} °F
6-K	Cr - 31 C - 1.6 Co - Bal		47 Rc	.296	188,000	325,000	32-1292 7.9
Star-J	Cr - 32 W - 17 Ni - 2.5	Fe - 3 C - 2.5 Co - Bal	Rc 60 1292-44	.316	75,000	335,000	32-1292 7.3
98-M2	Cr - 30 W - 18.5 V - 4 C - 2	Ni - 3.5 Fe - 2.5 Co - Bal	Rc 62.7 1292-45	.312	95,000	370,000	68-1292 7.2
Group IV							
J-1500 (M-252)	Cr - 20 Co - 10 Mo - 10 Ti - 3.0	C - .15 Al - 1.0 Ni - Bal	Rc 38 1200-33	.298	180,000		70-1200 7.15
Rene '41	Cr - 19 Co - 11 Mo - 10 Ti - 3.1	C - .09 Al - 1.5 Ni - Bal	Rc 41	.298	206,000		70.1200 7.8

Material	Chemical Composition %	Hardness at °F	Density lbs/in^3	Ultimate Tensile Strength	Compression Strength psi	Transverse Rupture Strength psi	Coefficient of Thermal Expansion $x\ 10^{-6}$ °F
Group V							
AD-99	99% Al_2O_3	Knoop 2000	.141	34,000	300,000		25-1000°C 9.2×10^{-6}
AP-100	100% Al_2O_3		.112		125,000		
Group VI							
CT-50	Tic - 80 Ni - 20	Ra 89	.198	100,000	500,000	150,000	300-1200 5.0
K-162B	TiC - 64 Ni - 25 Mo - 5 Cb - 4.5	Ra 89 1400-74	.217	112,000	450,000		5.3×10^{-6}
Group VII							
Pyroceram 9606		Knoop 700	.0939				
Group VIII							
Ferro-TiC	Ti - 26 C - 7 Cr - 2 Mo - 2 Fe - Bal	Rc 69 1200-50			360,000	200,000	-1200 5.2

TABLE III
PHASE II-A-1
TEST RESULTS

Test No.	Combination: Rolls	Combination: Cylinders	Coefficient of Friction, f: Initial	Max.	Final	Wear Inch	Change in Wt. gms. *	Wear Surfaces**
1	6-B	Rene' 41	.030	.030	.006	.0018	+.0013,+.0016	C
2	6-B	Rene' 41	.015	.015	.015	.0025	+.0015,+.0006	C
3	Rene' 41	CA-3	.045	.060	.033	.0013	-.0010,-.0013	C
4	Rene' 41	CA-3	.052	.082	.037	.0009	-.0008,-.0013	C
5	6-B	AD-99	.082	.082	.060	.0010	.0000,+.0001	B
6	6-B	AD-99	.037	.037	.003	.0011	+.0002, .0000	B
7	CA-3	6-B	.007	.007	.002	.0011	-.0005,-.0012	A
8	CA-3	6-B	.022	.022	.022	.0008	-.0003,-.0002	A
9	Star-J	608	.022	.062	.052	.0015	-.0006,+.0002	C
10	Star-J	608	.014	.045	.036	.0022	-.0013,+.0007	C
11	AD-99	K-162-B	.030	.046	.046	.0018	-.0001,+.0003	C
12	AD-99	K-162-B	.046	.053	.053	.0015	+.0002,-.0002	C
13	CA-3	6-B	.090	.135	.100	.0030	-.0001,-.0003	A
14	CA-3	6-B	.015	.015	.008	.0019	+.0004, .0000	A
15	AD-99	CA-3	.018	.018	.018	.0022	+.0002,-.0001	B
16	6-B	6-B	.004	.004	.001	.0021	+.0010,+.0008	A
17	K-162-B	K-162-B	.021	.021	.001	.0018	+.0003,+.0014	C
18	AD-99	CA-3	.001	.015	.008	.0030	-.0001,-.0001	B
19	Rene' 41	Rene' 41	.023	.029	.029	.0017	-.0010,-.0009	C
20	6-B	AD-99	.075	.090	.045	.0017	+.0008,+.0007	B
21	6-B	6-B	.020	.031	.031	.0016	+.0009,+.0010	A
22	AD-99	CA-3	.008	.014	.014	.0014	+.0012,+.0012	B
23	6-B	6-B	.010	.011	.011	.0022	+.0006,+.0007	A
24	AD-99	AD-99	.013	.013	.008	.0018	+.0005,+.0006	B

250,000 ft, 1200°F temp., 50 oscillation/min. (.50" stroke), 72,000 cycles, 200 lbs. load (150,000 psi)

*First figure refers to top roll, second to the bottom.

**
A - Good
B - Fair
C - Poor

TABLE IV
PHASE II-A-2
TEST RESULTS

Test No.	Combination	Load LBS.	Cycles Completed	Coefficient of Friction, f Initial	Max.	Final	Wear Inch	Change in Wt. gms. *
1	A	2000	1,200	.069	.204	---	.0048	+.0037,+.0034
2	A	1000	4,000	.051	.051	.024	.0087	+.0005, 000
3	A	500	72,000	.036	.036	.012	.0035	-.0001,-.0003
4	B	500	72,000	.027	.027	.018	.0017	+.0002,+.0003
5	C	500	72,000	.030	.030	.020	.0029	+.0001,+.0003
6	A	500	72,000	.048	.048	.030	.0041	+.0002, 000
7	C	500	72,000	.028	.028	.021	.0030	+.0001,+.0002
8	B	500	72,000	.023	.023	.023	.0019	000 ,+.0004
9	C	1000	45,000	.054	.288	.288	.0042	-.0029,-.0022
10	C	1000	72,000	.051	.054	.045	.0049	-.0016,-.0014
11	B	1000	1,400	(terminated due to trouble in test rig)				
12	(Roll design 'A' was used. Test terminated due to misalignment in the rig)							
13	B	1000	72,000	.054	.054	.030	.0052	-.0002,-.0007
14	B	1500	72,000	.080	.080	.070	.0043	-.0084,+.0034
15	C	1500	7,900	.090	.090	.060	.0054	+.0005,-.0611

250,000 ft, 1200°F, 50 oscillation/min. (.50" stroke)

*First figure refers to top roll, second to the bottom

Combination	Roll	Cylinder
A	CA-3	6-B
B	6-B	6-B
C	6-B	CA-3

Load, Lbs.	Stresses, psi
500	200,000
1000	260,000
1500	300,000
2000	325,000

TABLE V
PHASE II-B
TEST RESULTS

Test No.	Combination	Dry Film*	Coefficient of Friction, f Initial	Max.	Final	Wear Inch
1	C	CaF_2	.036	.048	.048	.0026
2	B	CaF_2	.048	.048	.036	.0011
3	C	811	.035	.035	.018	.0012
4	B	811	.030	.033	.033	.0021
5	C	CaF_2	.051	.051	.038	.0019
6	B	CaF_2	.054	.054	.049	.0026
7	B	811	.037	.054	.042	.0020
8	C	811	.042	.042	.035	----
9	C	100-1	.033	.054	.054	.0019
10	B	100-1	.039	.053	.051	.0025
11	B	100-1	.052	.065	.065	.0035
12	C	100-1	.060	.060	.030	.0040
13	B	CDL-5940	.041	.047	.030	.0018
14	C	CDL-5940	.048	.049	.049	.0035
15	B	CDL-5940	.048	---	---	.0016
16	C	CDL-5940	.066	.066	.051	----

500 lbs. load (200,000 psi), 250,000 ft, 1200°F, 72,000 cycles, 50 oscillation/min. (.50" stroke)

*Dry film only on the rolls

Combination	Roll	Cylinder
B	6-B	6-B
C	6-B	CA-3

TABLE VI- Dimensional Changes in Rolls (before and after coating and tests) - PHASE II-B

Coating	Roll No.	Thickness of Coating Inch	Change in Wt. after Coating gms.	Surface Finish, RMS Micro In. Before Coating	After Coating	After Test	Change in Wt.* After Test gms.	Remarks
811	1	.00045	-.0030**	5	13	14	----	Slight metal welding near corners
	2	.00055	-.0028	6	12	15	-.0036	" " " " "
	3	.00070	-.0032	5	15	22	-.0006	High metal welding near corners
	4	.00070	-.0035	6	14	15	-.0009	" " " " "
	5	.00055	-.0043	5	15	20	-.0006	" " " " "
	6	.00065	-.0050	5	15	19	-.0008	" " " " "
	7	.00035	-.0057	5	13	18	-.0005	Slight metal welding near corners
	8	.00070	-.0045	6	16	11	-.0009	" " " " "
	9	.00075	-.0039	4.5	20			
	10	.00050	-.0051	4	12			
100-1	11	no	-.0004**	5	7	17	-.0021	Show better surface than 811
	12		-.0016	5	8	13	-.0014	" " " " "
	13		-.0031	6	7	19	-.0018	" " " " "
	14		-.0009	5.5	10	20	-.0020	" " " " "
	15	appreciable	-.0029	5	7	14	-.0019	" " " " "
	16		-.0024	5	8	15	-.0010	" " " " "
	17		-.0016	5	8	16	-.0021	" " " " "
	18		-.0018	5	8	22	-.0012	
	19	thickness	-.0014	5	7			
	20		-.0031	5	8			
CaF_2	21	.0015	+.0143	5.5	165			
	22	.0011	+.0125	5	170	20	+.0011	No metal welding
	23	.0014	+.0108	5.5	165	21	-.0048	
	24	.0013	+.0135	5	165	19	-.0048	
	25	.0010	+.0145	5	120	22	-.0054	Coating removed
	26	.0014	+.0194	6	160	19	-.0070	
	27	.0010	+.0232	7	120	19	-.0063	
	28	.0012	+.0206	5	165	18	-.0051	Smooth surfaces
	29	.0008	+.0164	5.5	165	22	-.0052	
	30.	.0012	+.0135	5	125			
CDL-5940	31	no	+.0009	7		21	-.0029	Very little metal to metal welding or pitting
	32		+.0006	5	9	18	-.0036	
	33		+.0008	5		15	-.0023	" " " "
	34		+.0009	5		17	-.0016	
	35	appreciable	+.0004	5	to	21	-.0018	" " " "
	36		+.0009	5		24	-.0031	
	37		+.0010	4.5		16	-.0011	" " " "
	38		+.0005	5		20	-.0007	
	39	thickness	+.0005	6	9.5	18	-.0012	" " " "
	40		+.0004	6		23	-.0014	

*: Difference taken between the weights of After Coating and After Test.
**: Decrease in weight after coating with 811/ could be due to the sand blasting procedure followed prior to coating. and 100-1

TABLE VII PHASE III TEST RESULTS

Test Number	1	2	3	4	5	6	7
Bearing	BR-16-5	BR-16-5	BR-16-5	BR-16-5	BR-16-5	BR-16-5	BR-16-5
Bearing Number*	24 8	12 14	7 17	20 22	5 28	28 5	28 5
Oscillation, cycles @100 osc/min $\pm 10°$	40,000	40,000	20,000	18,000	1,590	40,000	40,000
Load, lbs.	2000	2000	2000	2000	45	45	2000
Stresses, psi	175,000	175,000	175,000	175,000	-------	-------	175,000
Temperature, °F	1200	1200	1200	1200	room	room	room
Environment, mm Hg	.020	.020	.020	.020	760	760	760
Initial R. C.	.0035	.0035	.005	tight	.0035	.0035	.0035
Final R. C.	locked	locked	locked	tight	tight	.0035	.0035
Lubricant	none	none	none	CaF_2 Oxide Frit	none	oil	oil
Remarks:	destroyed	destroyed	destroyed	destroyed	**	satisfactory Performance	satisfactory performance

*First number refers to top bearing second number, to bottom bearing

**Stopped before destruction for use in Tests 6 & 7

TABLE VII (CONTINUED)

Test Number	8	9	10	11	12	13	14
Bearing	BR-16-5	BR-16-5	BR-16-5	BR-16-5	BR-16-5	BR-16-5	BR-16-5
Bearing Number*	5 28	5 28	11 15	11 15	11 15	1 2	1 2
Oscillation, cycles @100 osc/min $\pm 10°$	8,000	1,500	6,000	40,000	15,000	6,000	20,000
Load, lbs.	45 & 500	45 & 500	45 & 500	45 & 500	2000	45	500 & 2000
Stresses, psi	110,000	110,000	110,000	110,000	175,000	-------	175,000
Temperature, °F	room	1200	room	1200	1200	room	1200
Environment, mm Hg	760	.020	760	.020	.020	760	.02
Initial R. C.	.0035	.0035	.0035	.0035	----	.0035	.0035
Final R. C.	.0035	locked	.0035	000	locked	.0035	tight
Lubricant	none	none	none	none	none	none	none
Remarks:	satisfactory performance	destroyed badly locked	satisfactory performance	**	destroyed badly locked	satisfactory performance	One of the cages broken; High rubbing

*First number refers to top bearing second number, to bottom bearing

**Tight, bearing free after little tapping

TABLE VII (CONTINUED)

Test Number	15	16	17	18	19	20	21
Bearing	BR-16-5	BR-16-5	BR-16-5	BR-16-5	BR-16-5	BR-16-5	BR-16-5
Bearing Number*	25 3	25 3	3 25	9 23	10 12	6 26	13 19
Oscillation, cycles @100 osc/min ±10°	6000	40,000	15,400	19,634	40,000	40,000	40,000
Load, lbs.	45	500 & 2000	5000	5000	2000	2000	5000
Stresses, psi	-------	200,000	280,000	280,000	200,000	200,000	280,000
Temperature, °F	room	1200	1200	1200	1200	1200	1200
Environment, mm Hg	760	.02	.02	.02	.02	.02	.02
Initial R. C.	.0040	.004	---	.0045	.0002	.0008	---
Final R. C.	.0040	little tight	---	tight	tight	free	free
Lubricant	none	none	none	none	CaF_2	MRC DF-700 (Cage & rolls)	MRC DF-700 (All surfaces)
Remarks:	Satisfactory performance	Satisfactory however, rough surfaces	Rig failure; high skewing on races	High torque & vibration	Little tight rolls in bad shape	**	***

*First number refers to top bearing, second number - to bottom bearing

**Best condition; still metal-to-metal contact after about 20,000 cycles.

***Vibration & high torque at the final period of test.

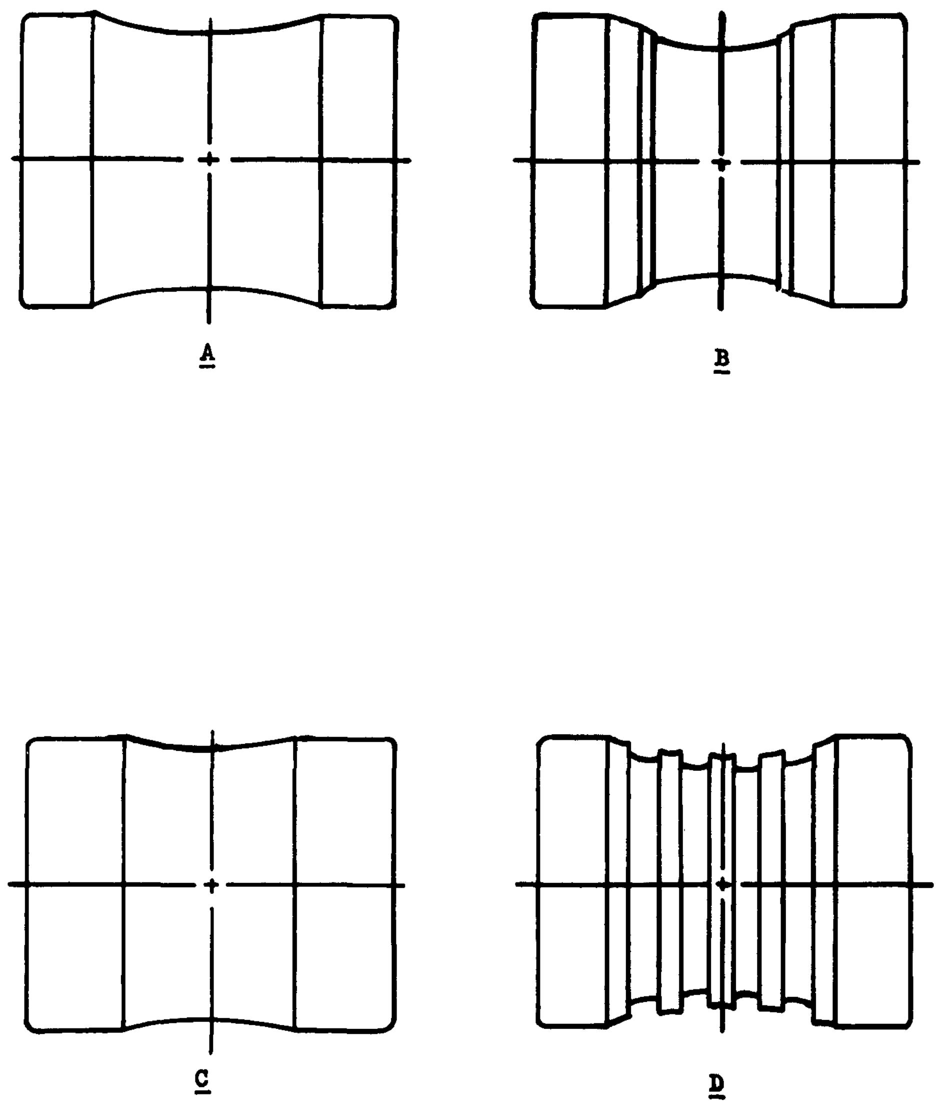


FIGURE 1, FOUR ROLL CONFIGURATION

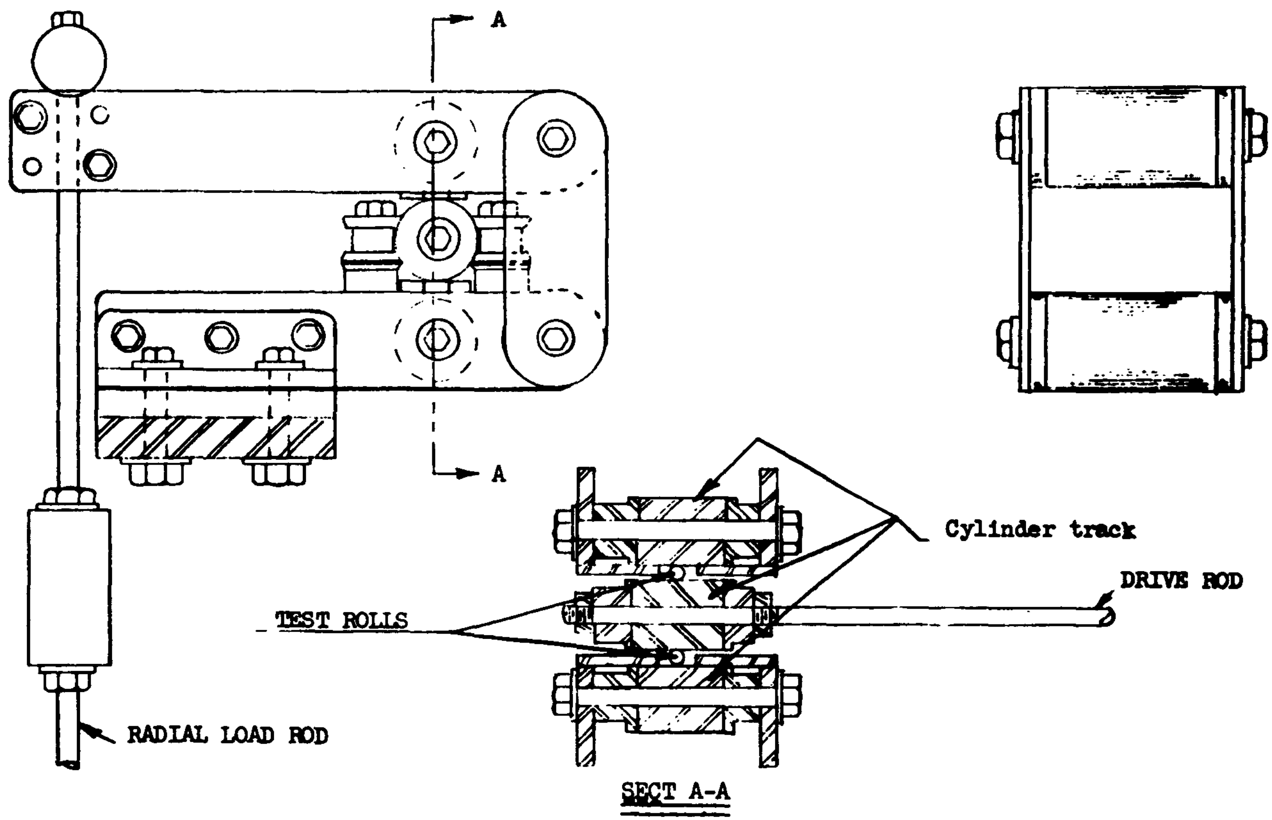


FIGURE 2, TEST RIG FOR ROLL SPECIMENS EVALUATION

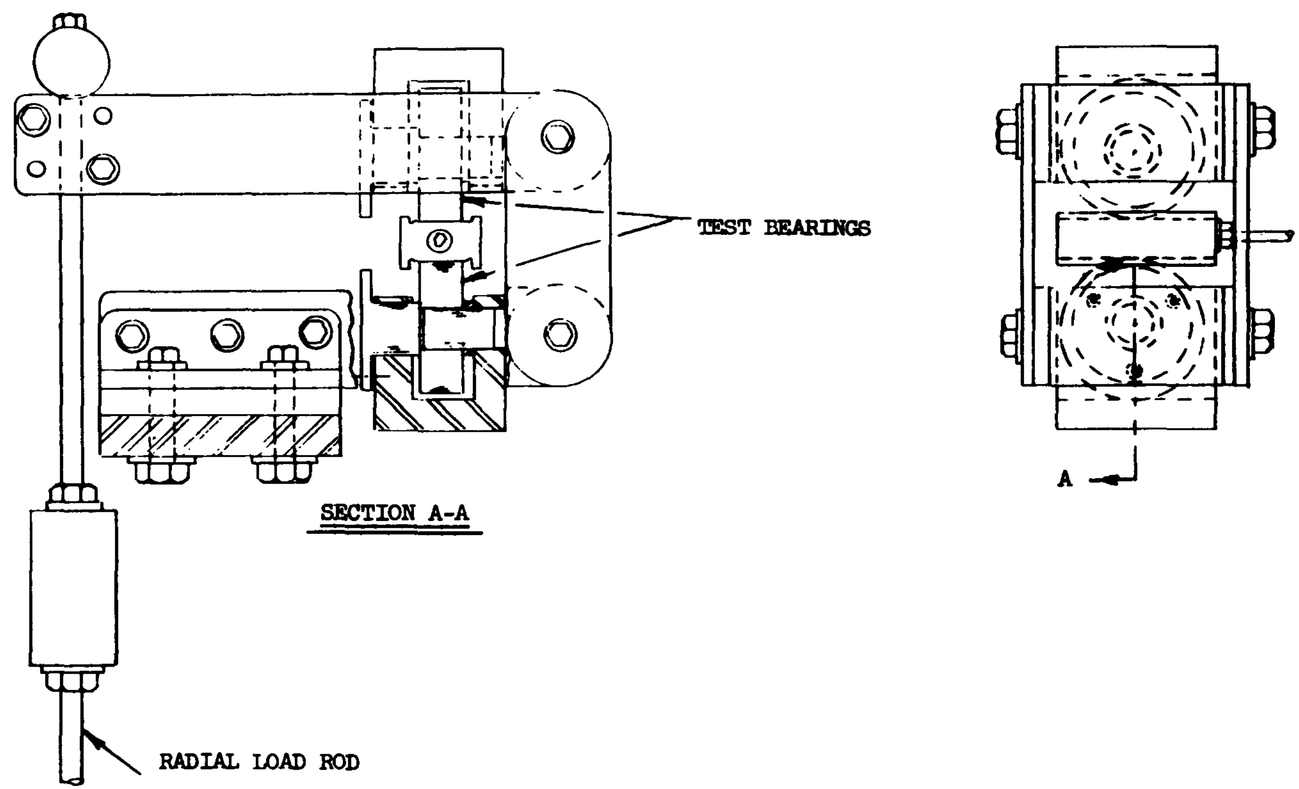


FIGURE 3, TEST RIG FOR BEARING EVALUATION

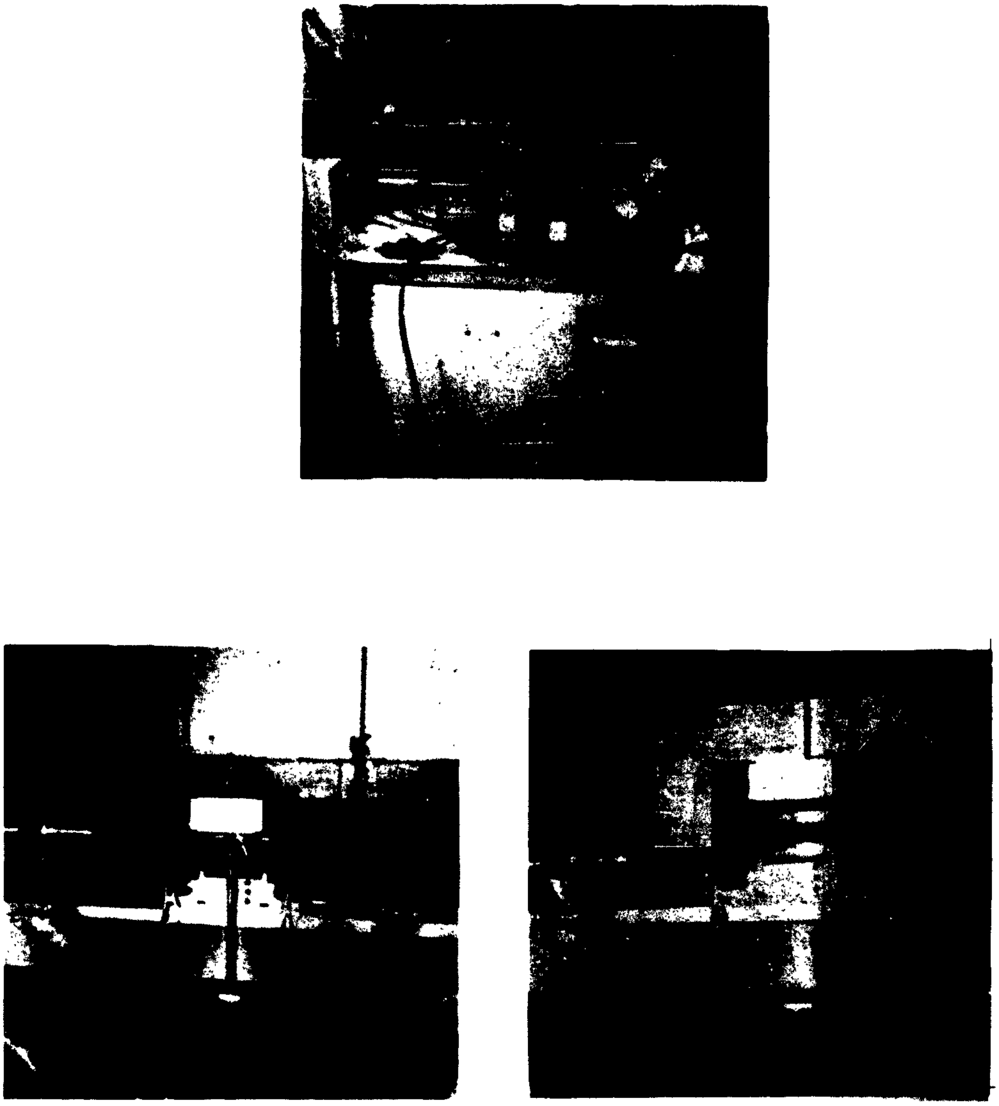
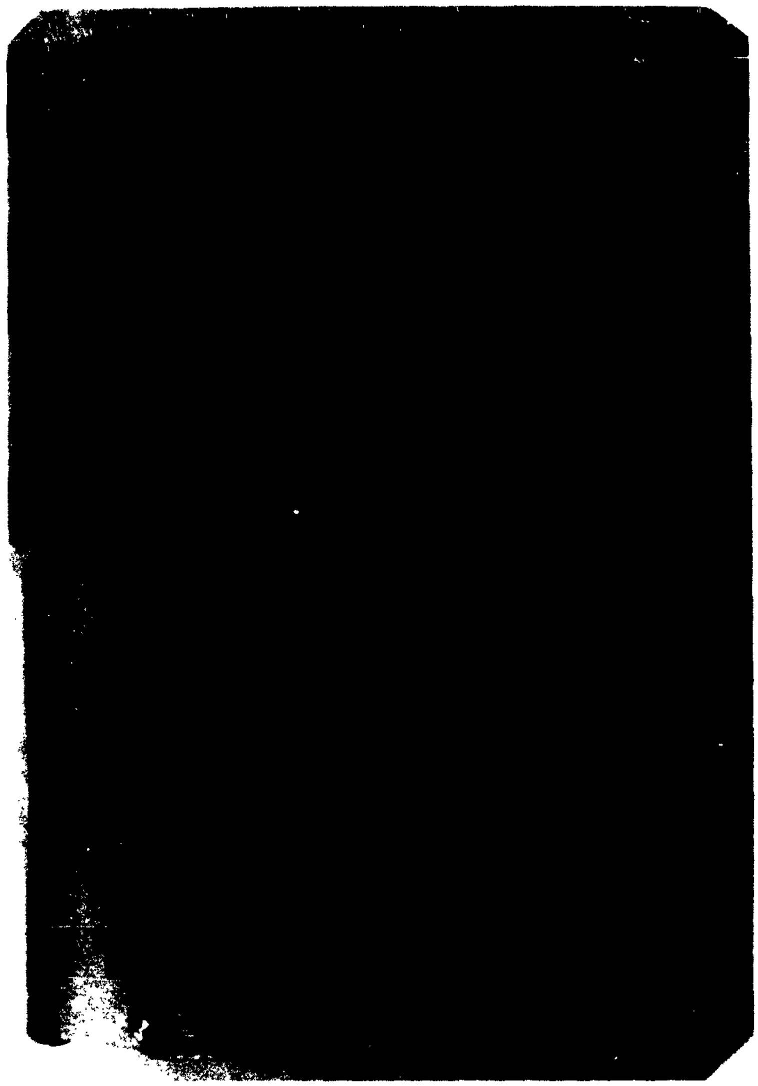


FIGURE 4, TEST FACILITY

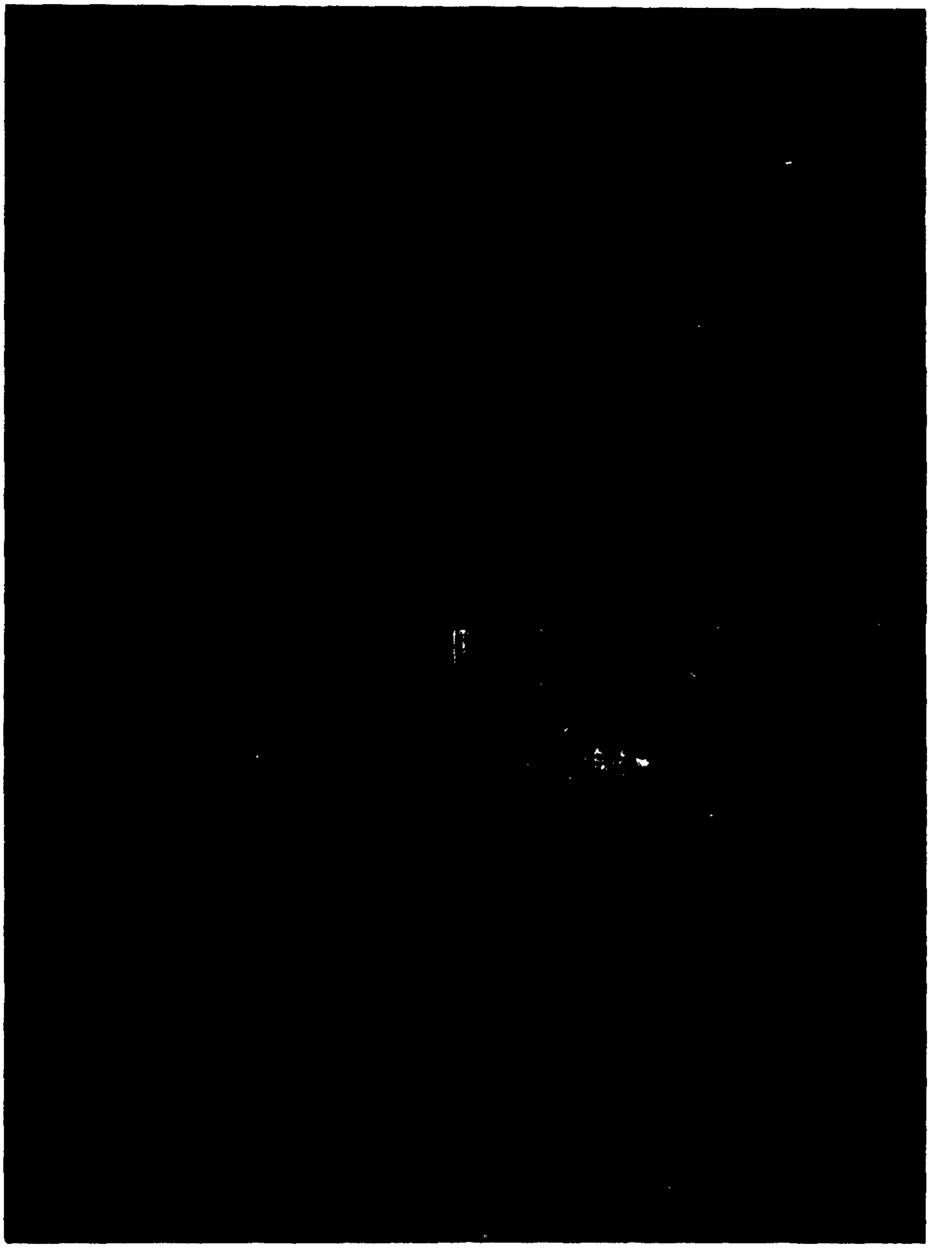
A D



C B

FIGURE 5, DIFFERENT ROLLS AFTER INVESTIGATION

A D



C B

FIGURE 6, DIFFERENT CYLINDERS AFTER INVESTIGATION

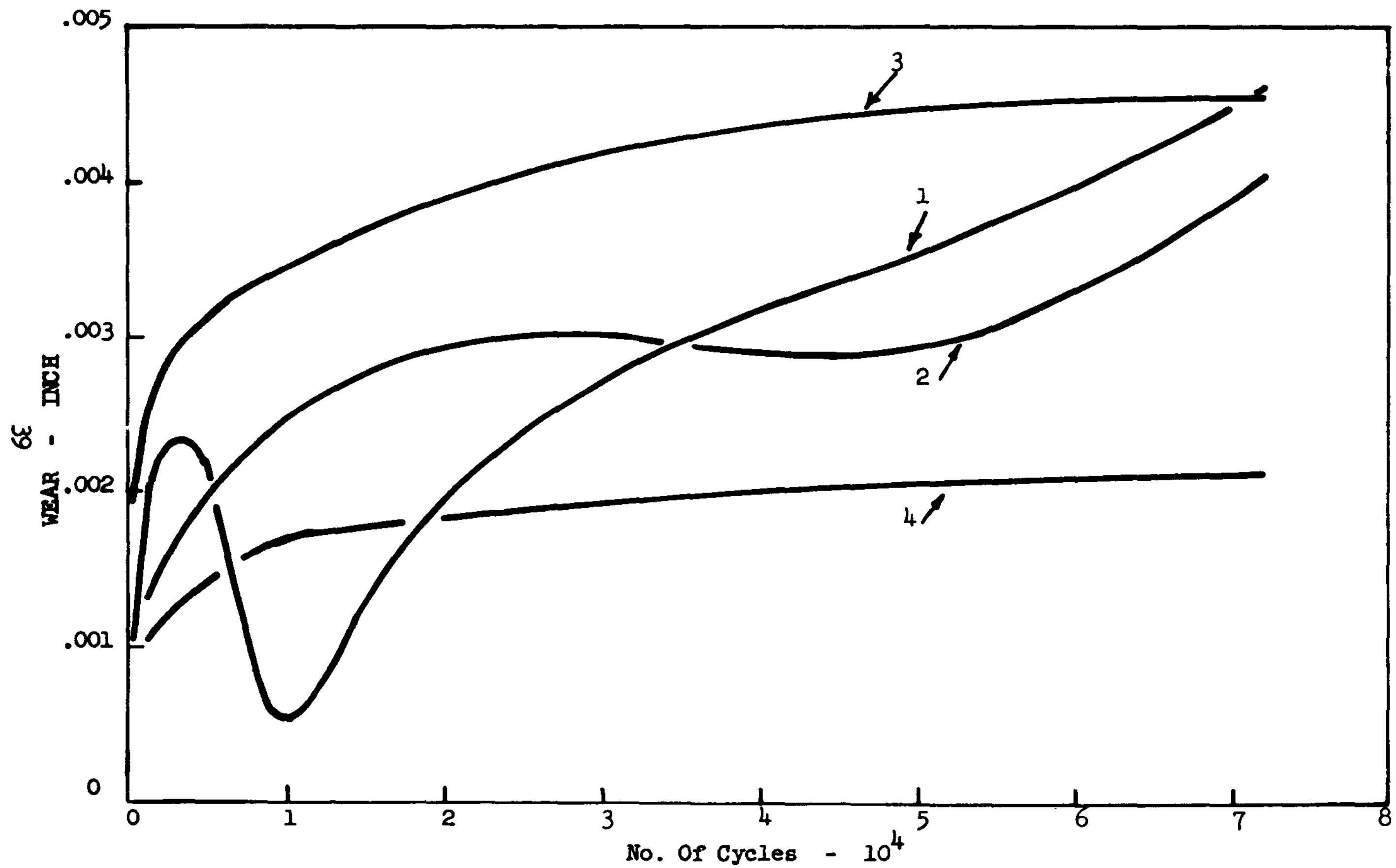


FIGURE 7, NO. OF CYCLES VS WEAR FOR TESTS 1, 2, 3 & 4 (PHASE I)

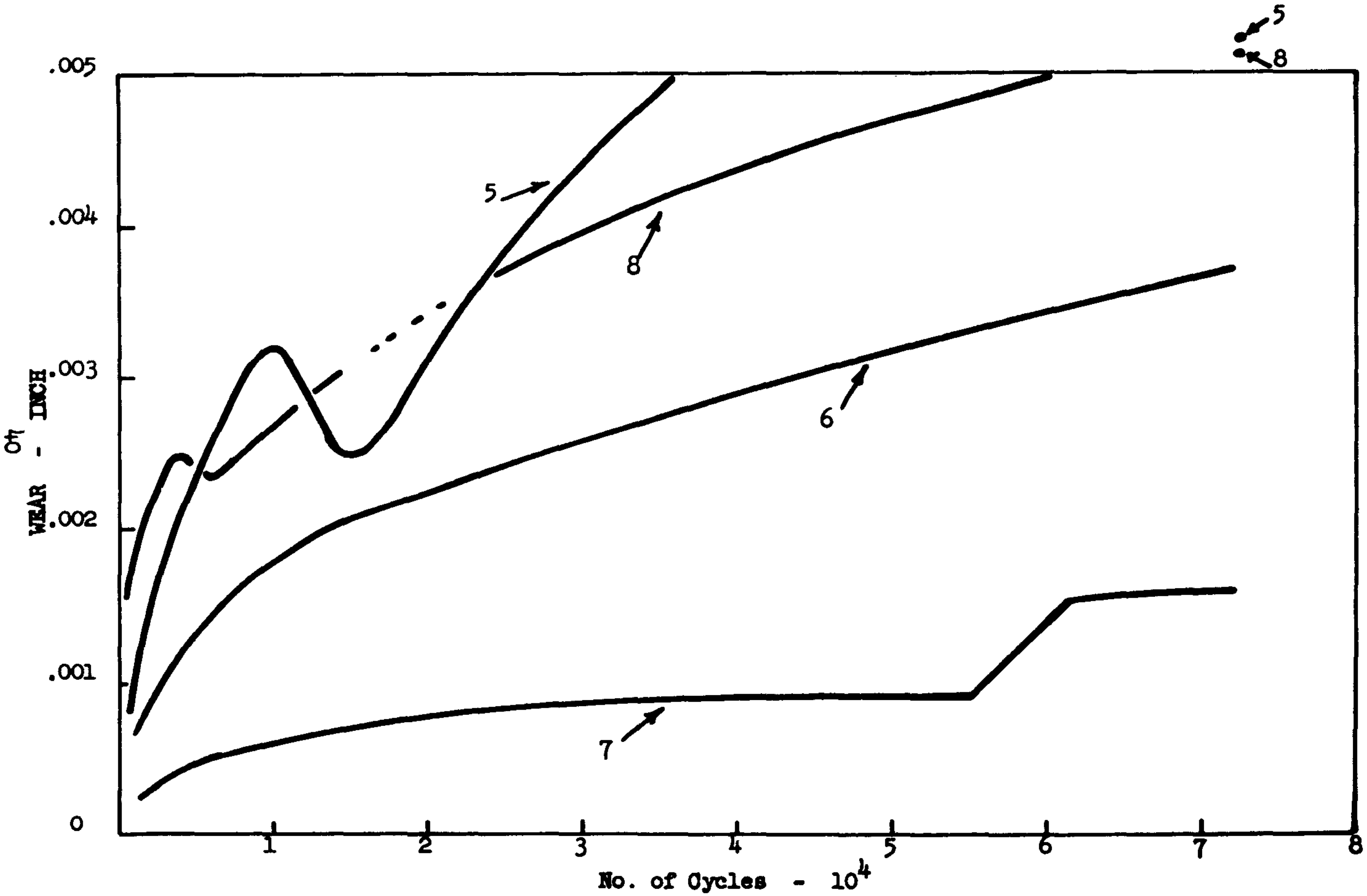


FIGURE 8, NO. OF CYCLES VS WEAR FOR TESTS 5, 6, 7 & 8 (PHASE I)

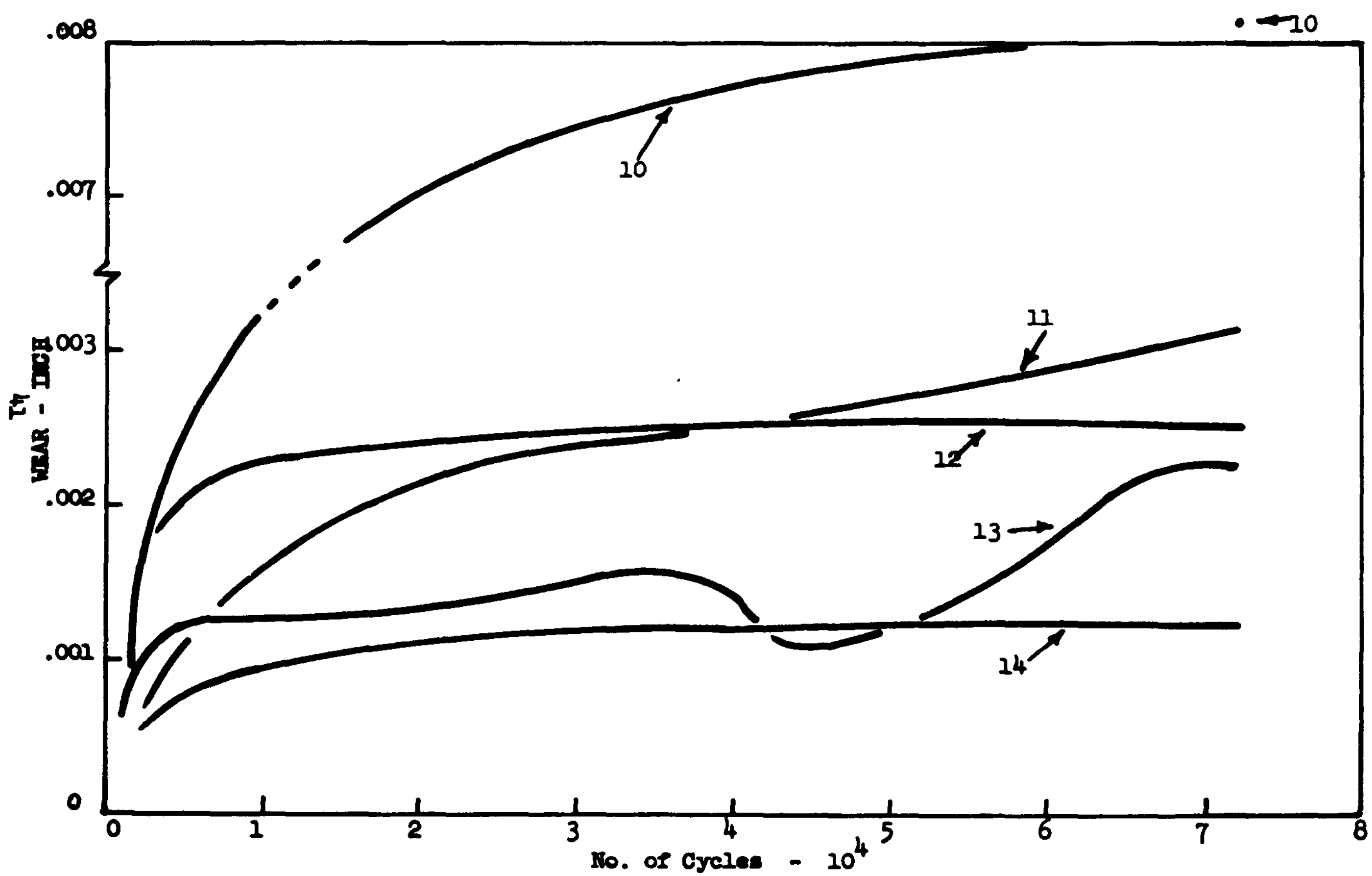


FIGURE 9, NO. OF CYCLES VS WEAR FOR TESTS 10, 11, 12, 13 & 14 (PHASE I)

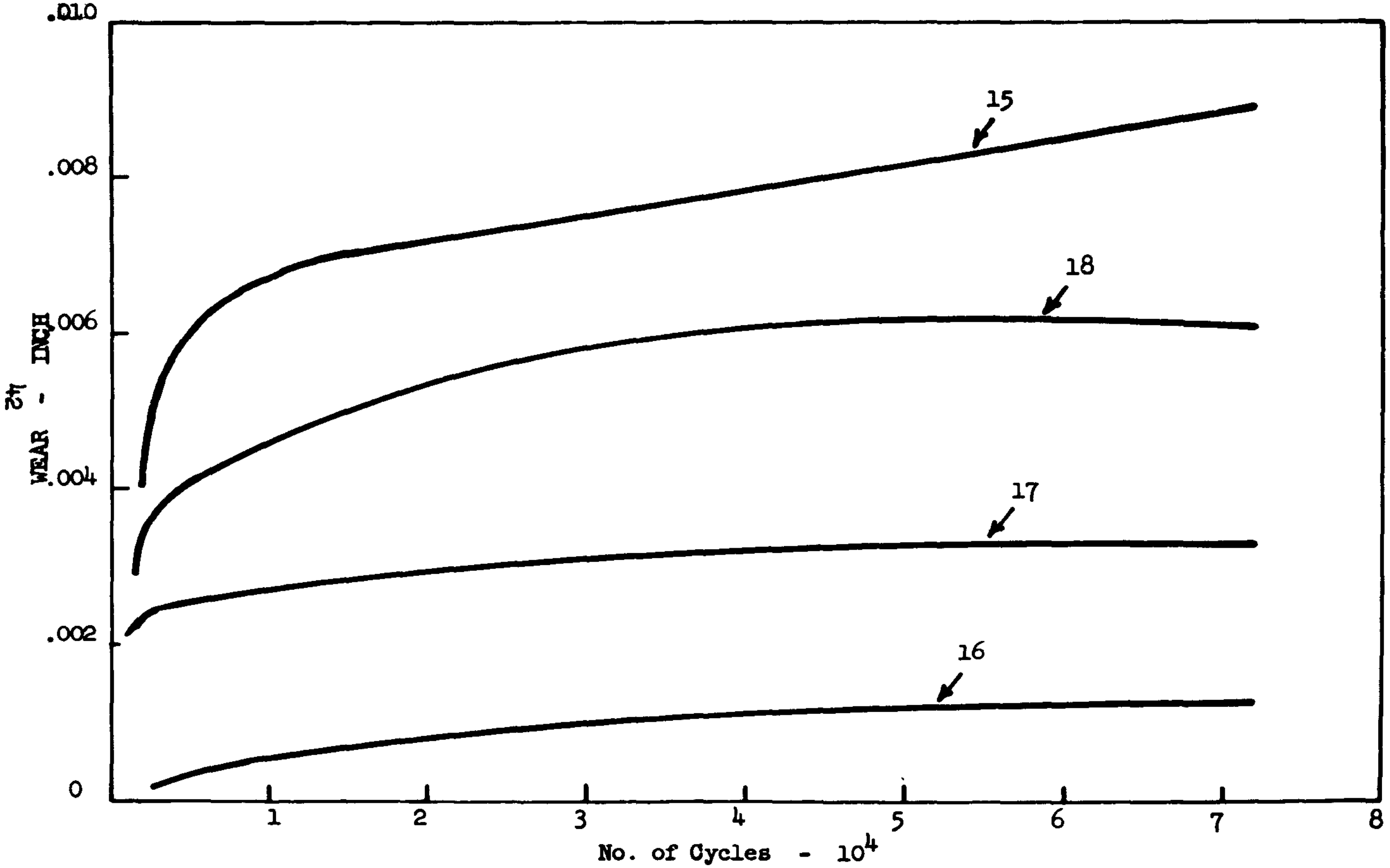


FIGURE 10, NO. OF CYCLES VS WEAR FOR TESTS 15, 16, 17 & 18 (PHASE I)

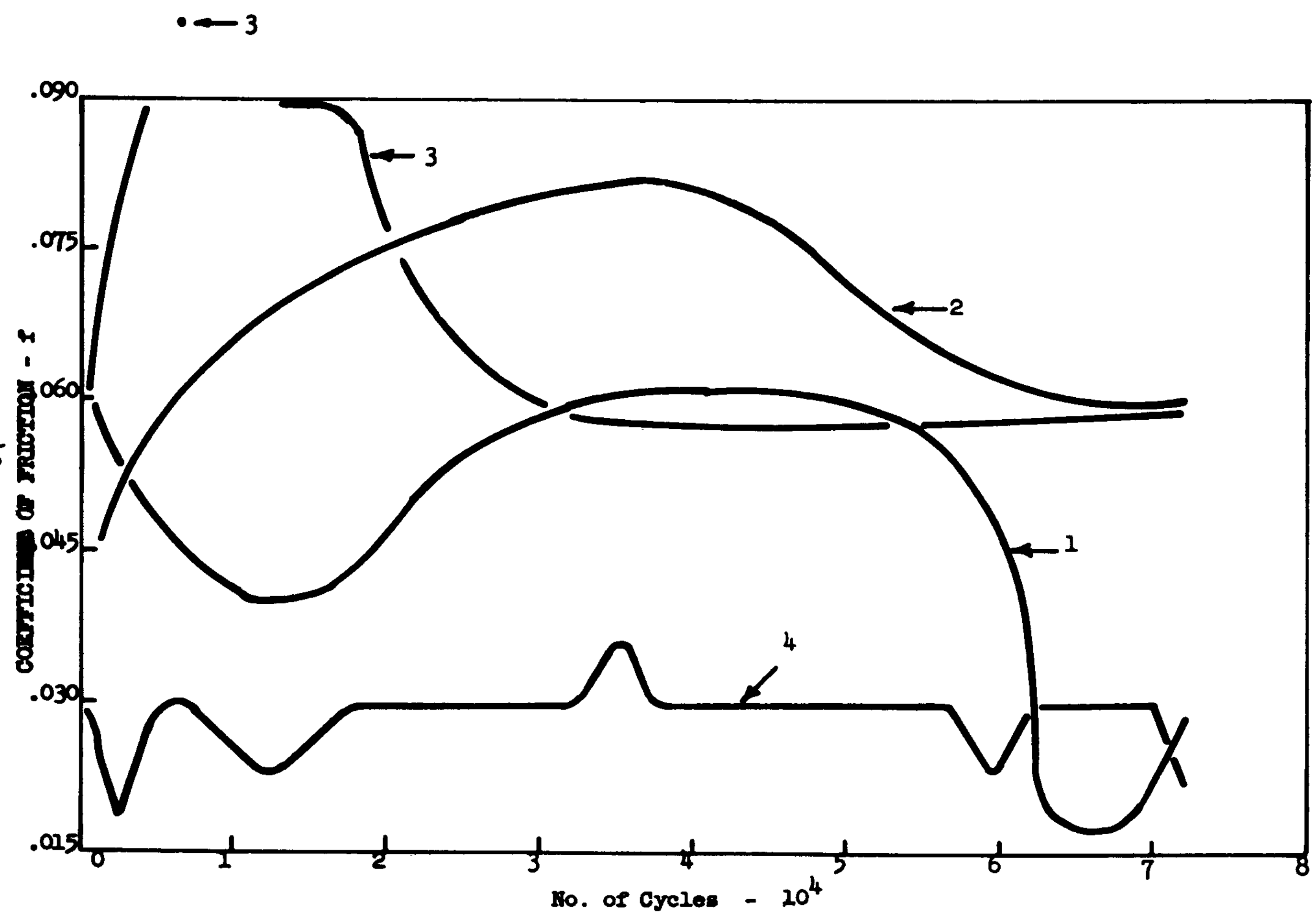


FIGURE 11, NO. OF CYCLES VS FRICTION FOR TESTS 1, 2, 3 & 4 (PHASE I)

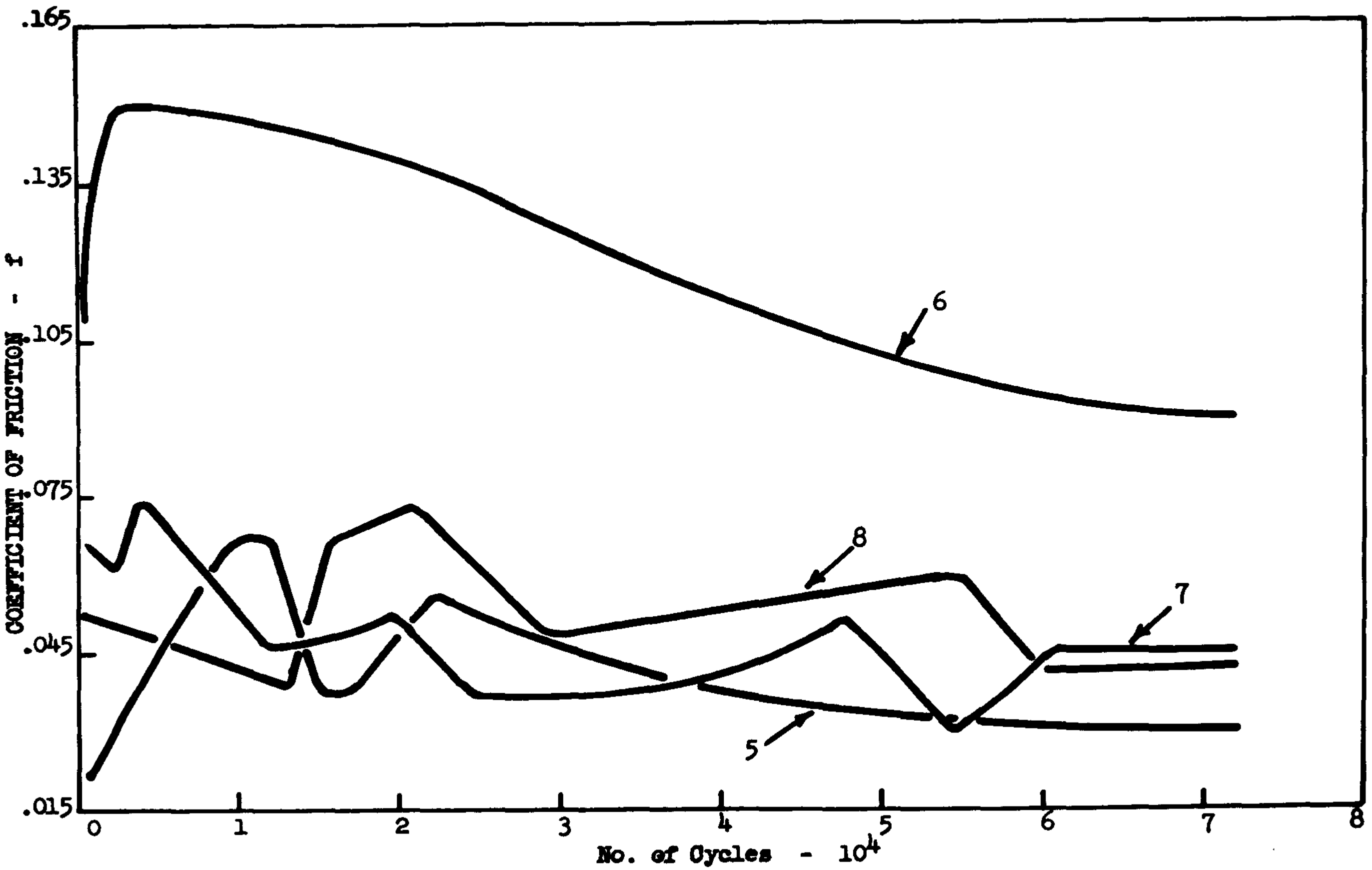


FIGURE 12, NO. OF CYCLES VS FRICTION FOR TESTS 5, 6, 7 & 8 (PHASE I)

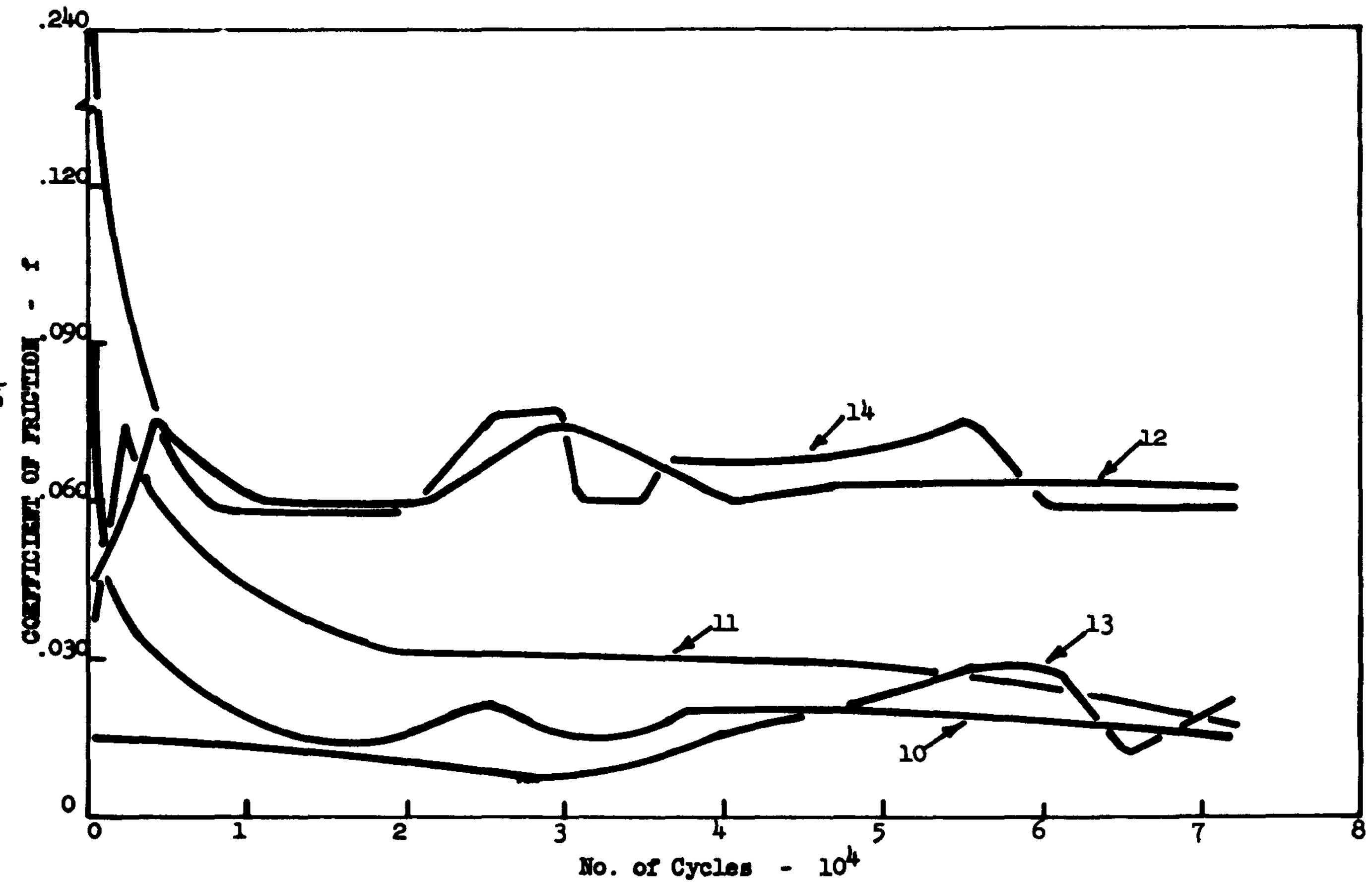


FIGURE 13, NO. OF CYCLES VS FRICTION FOR TESTS 10, 11, 12, 13 & 14 (PHASE I)

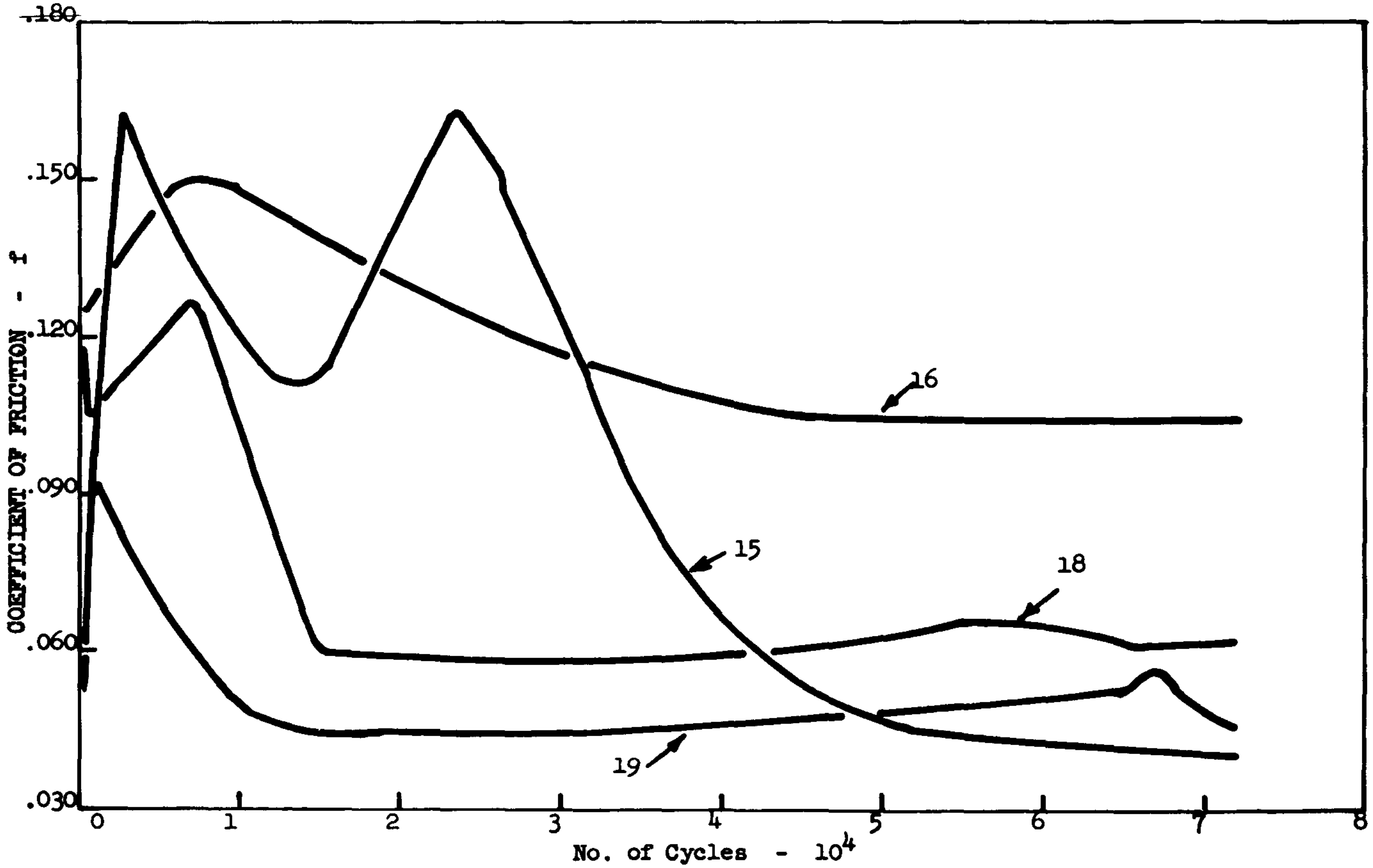


FIGURE 14, NO. OF CYCLES VS FRICTION FOR TESTS 15, 16, 17 & 18 (PHASE I)

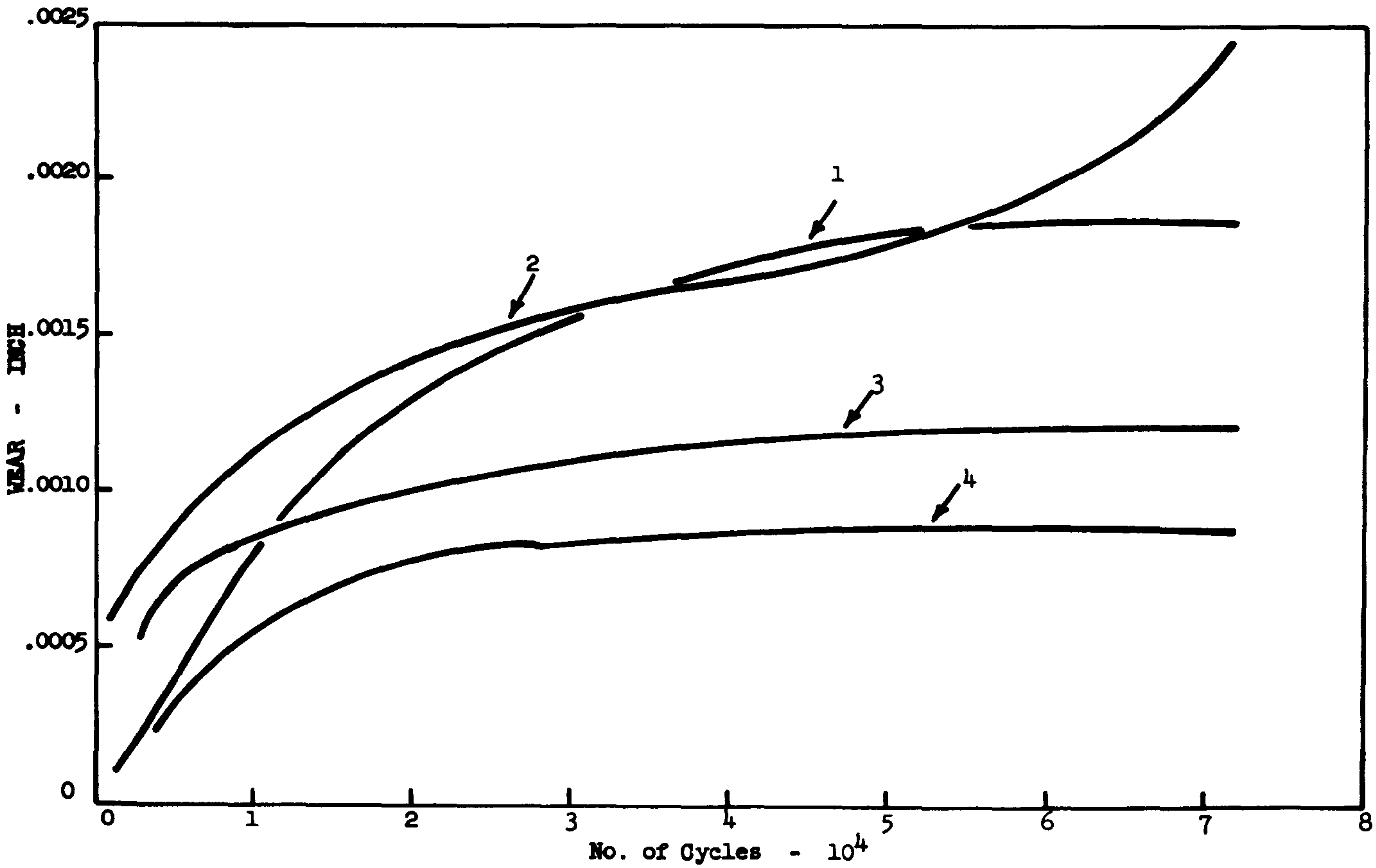


FIGURE 15, NO. OF CYCLES VS WEAR FOR TESTS 1, 2, 3 & 4 (PHASE II-A-1)

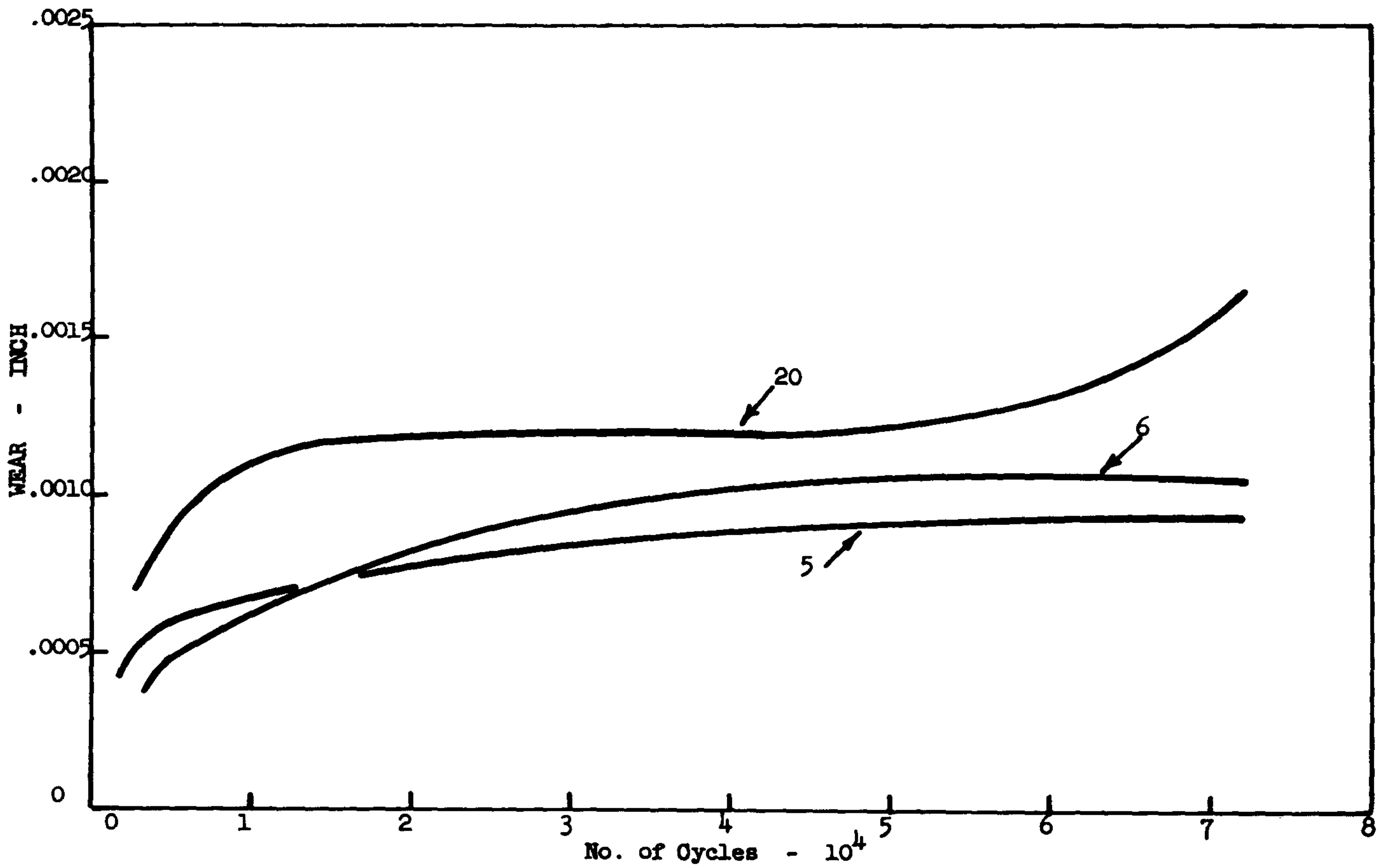


FIGURE 16, NO. OF CYCLES VS WEAR FOR TESTS 5, 6 & 20 (PHASE II-A-1)

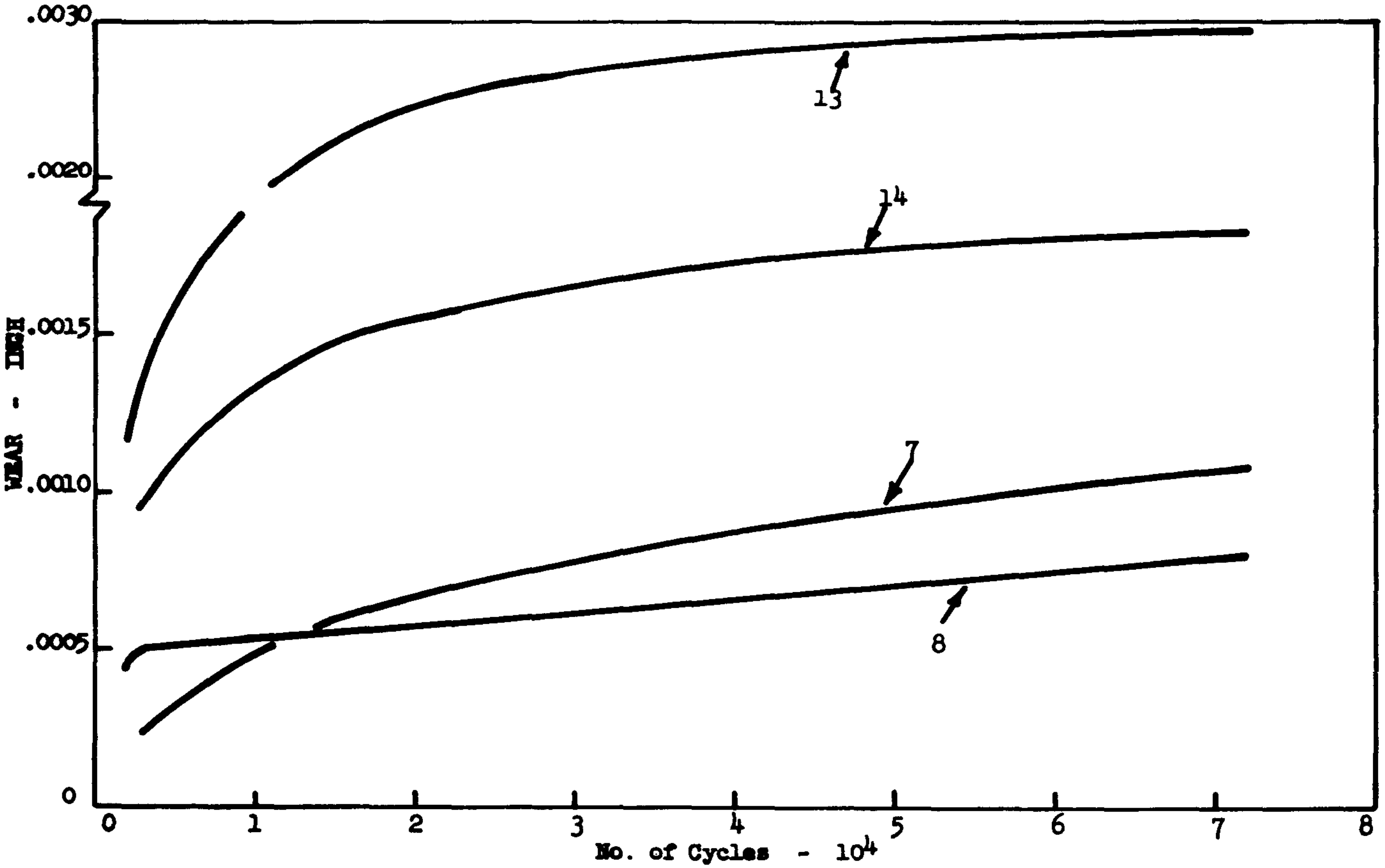


FIGURE 17, NO. OF CYCLES VS WEAR FOR TESTS 7, 8, 13 & 14 (PHASE II-A-1)

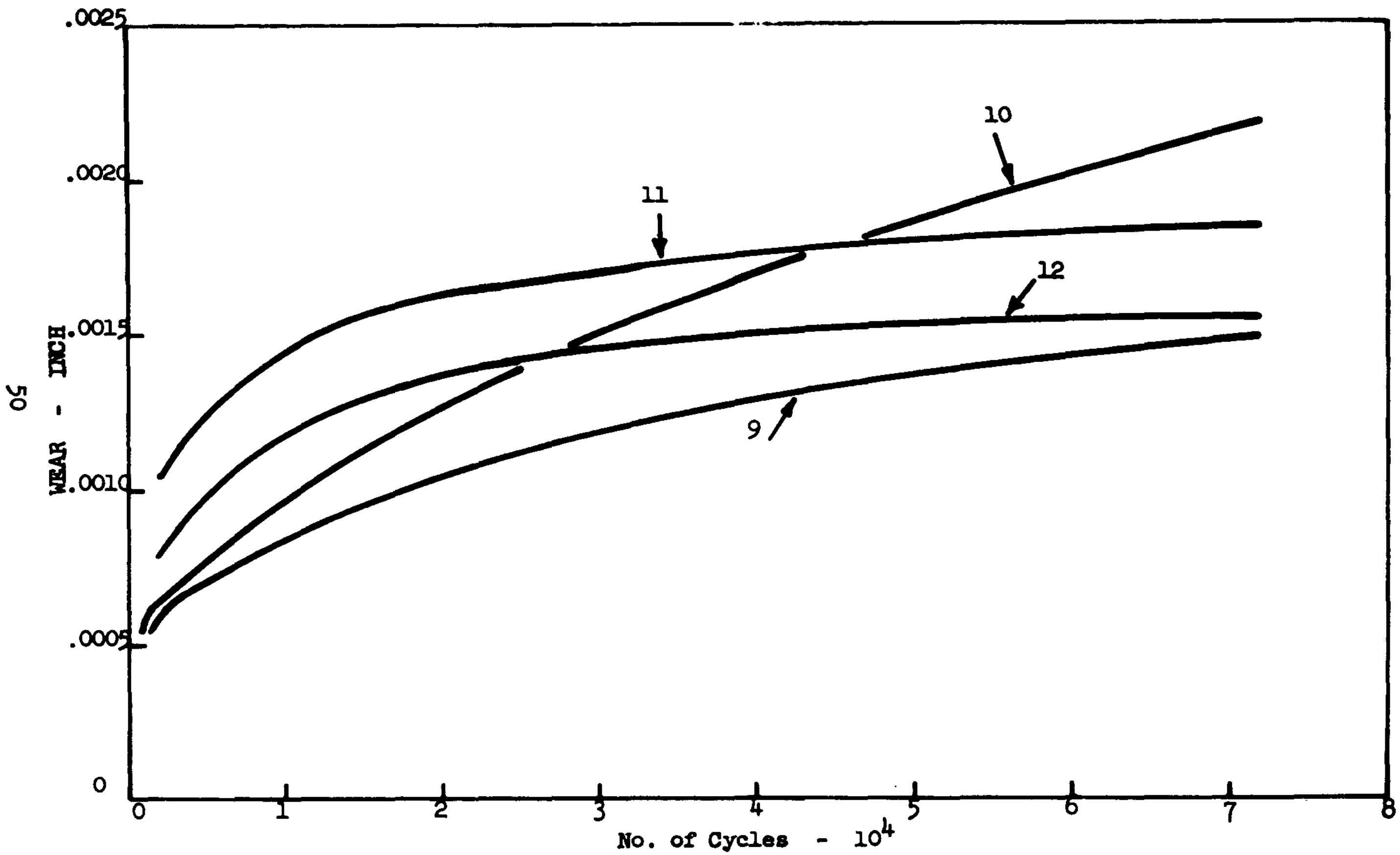


FIGURE 18, NO. OF CYCLES VS WEAR TESTS FOR 9, 10, 11 & 12 (PHASE II-A-1)

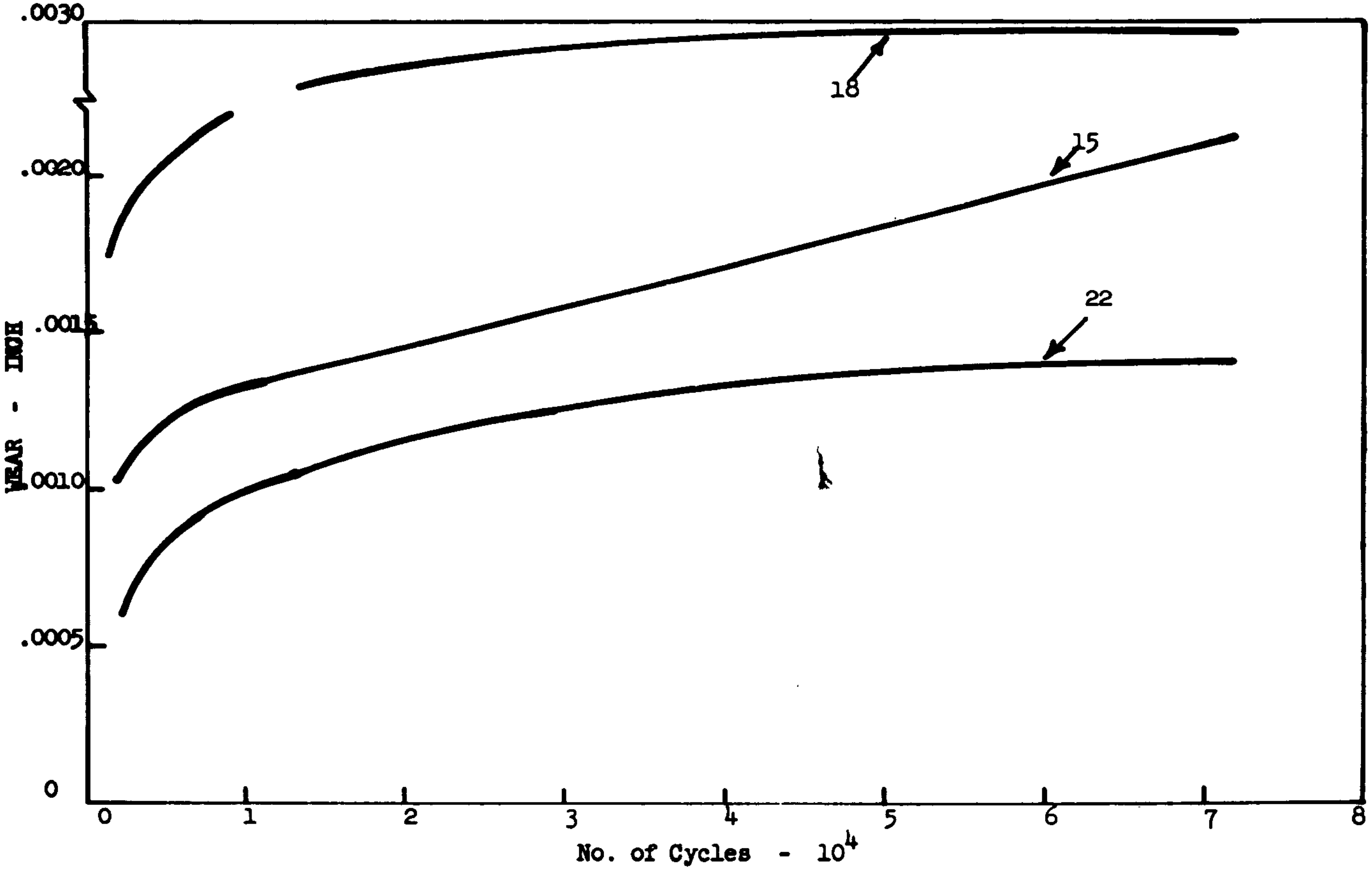


FIGURE 19, NO. OF CYCLES VS WEAR FOR TESTS 15, 18 & 22 (PHASE II-A-1)

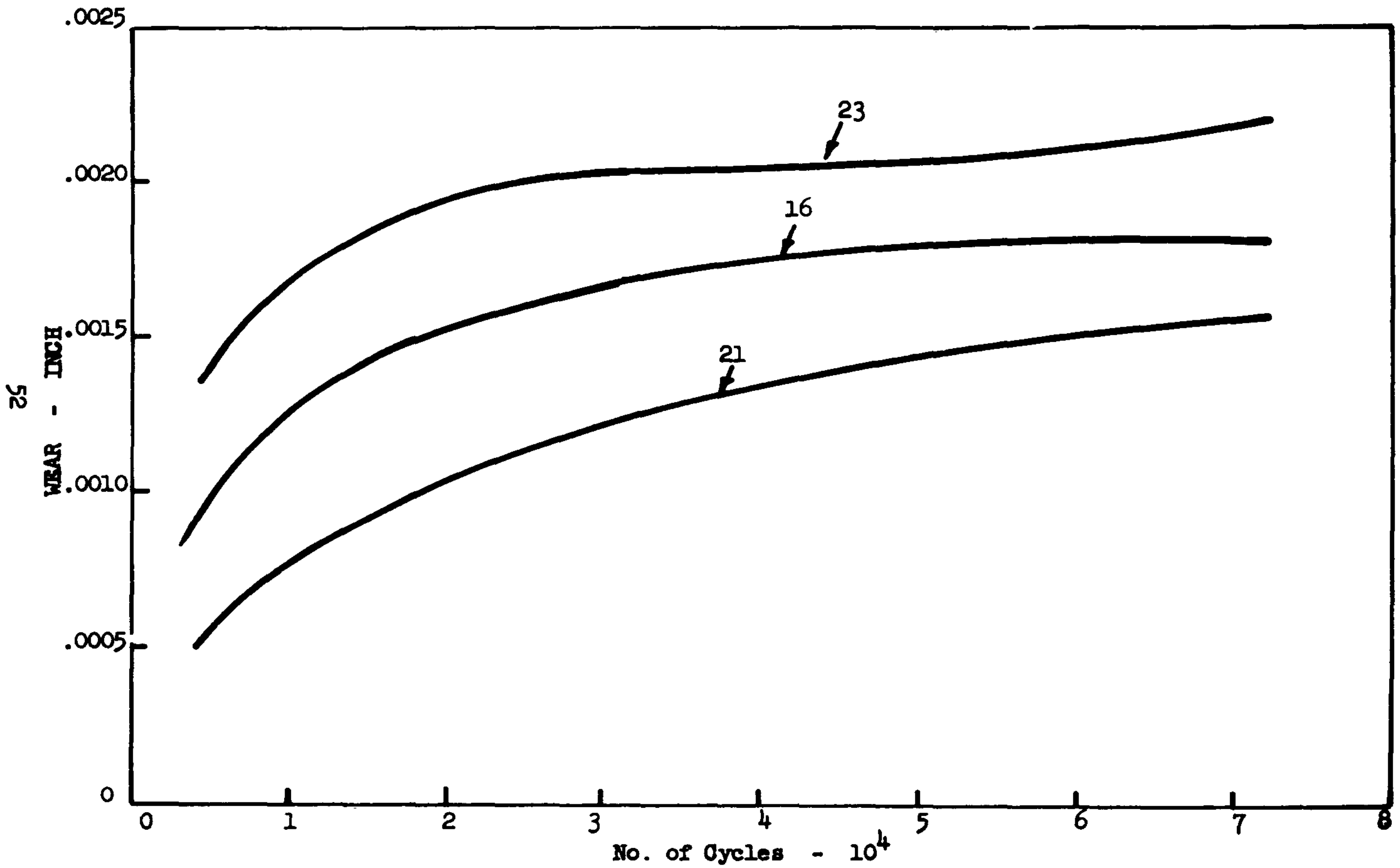


FIGURE 20, NO. OF CYCLES VS WEAR FOR TESTS 16, 21 & 23 (PHASE II-A-1)

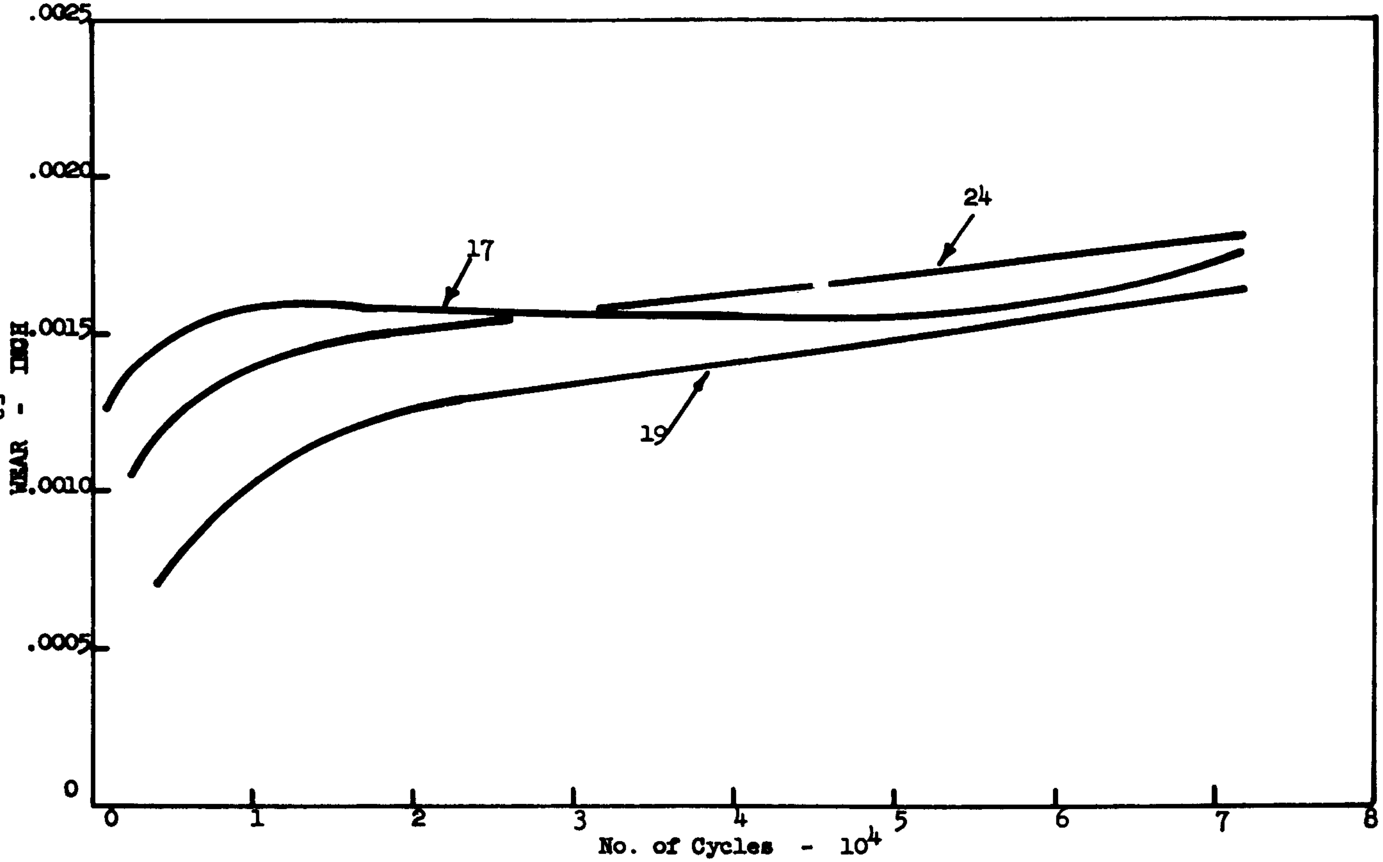


FIGURE 21, NO. OF CYCLES VS WEAR FOR TESTS 17, 19 & 24 (PHASE II-A-1)

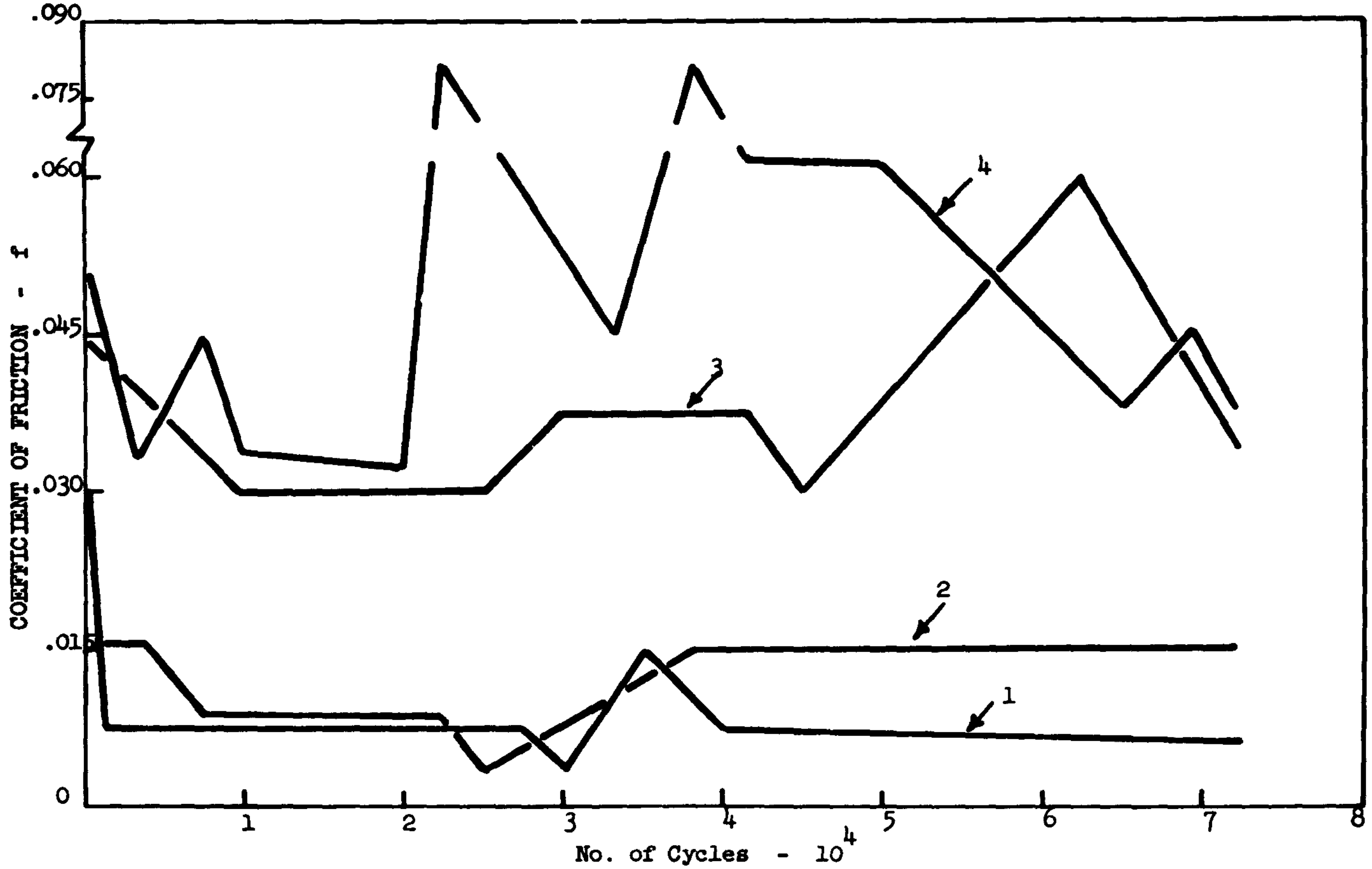


FIGURE 22, NO. OF CYCLES VS FRICTION FOR TESTS 1, 2, 3, & 4 (PHASE II-A-1)

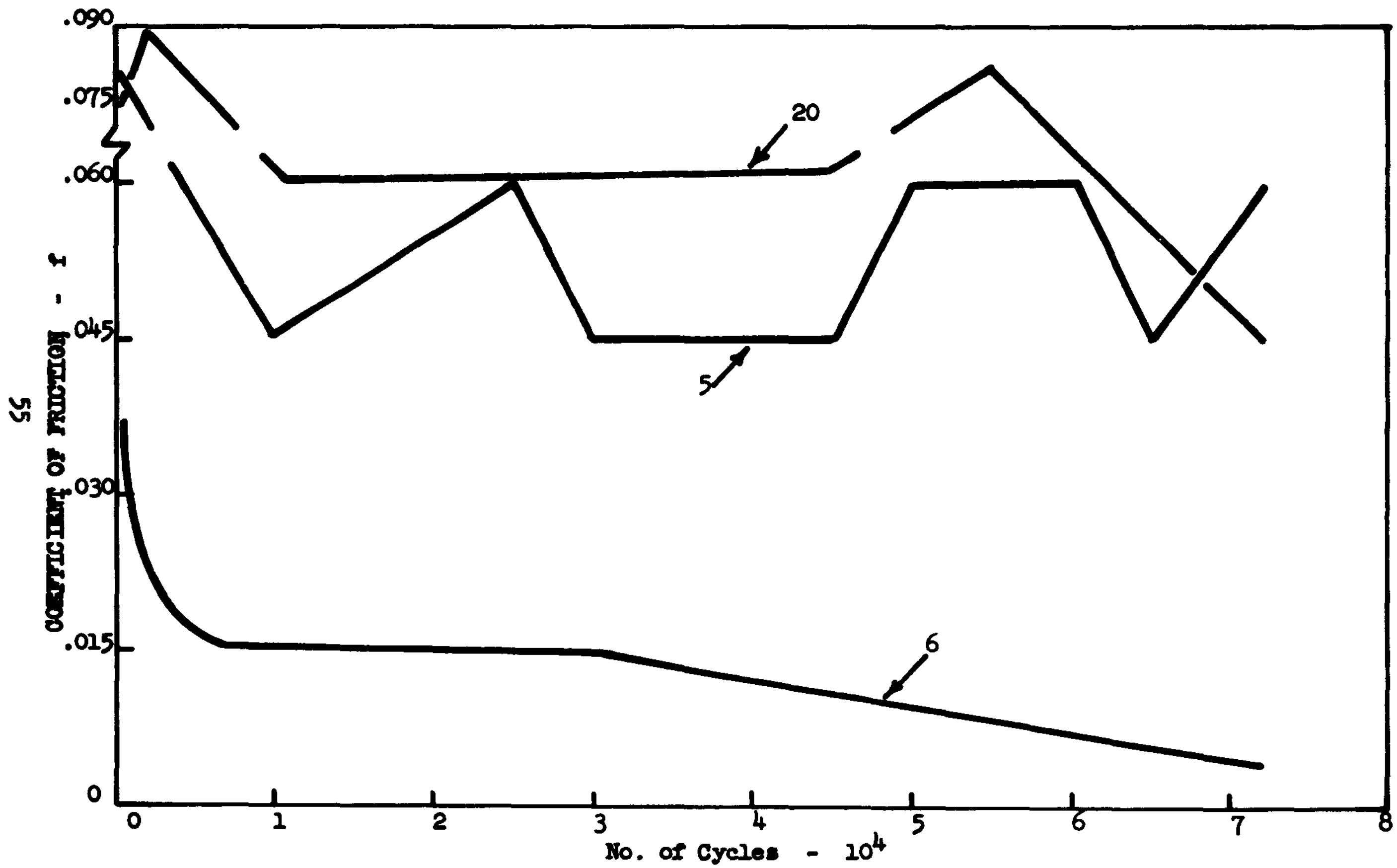


FIGURE 23, NO. OF CYCLES VS FRICTION FOR TESTS 5, 6 & 20 (PHASE II-A-1)

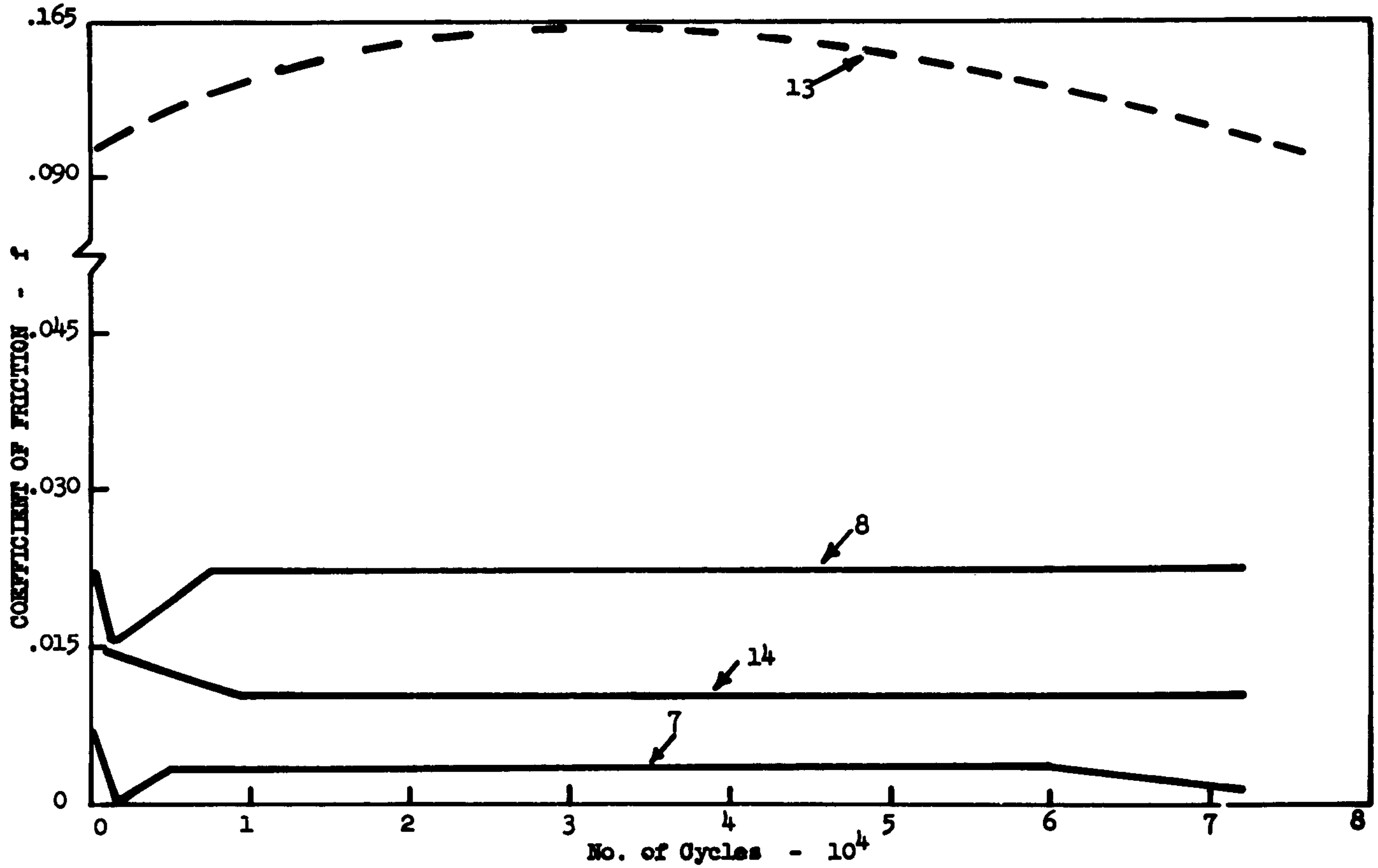


FIGURE 24, No. OF CYCLES VS FRICTION FOR TESTS 7, 8, 13 & 14 (PHASE II-A-1)

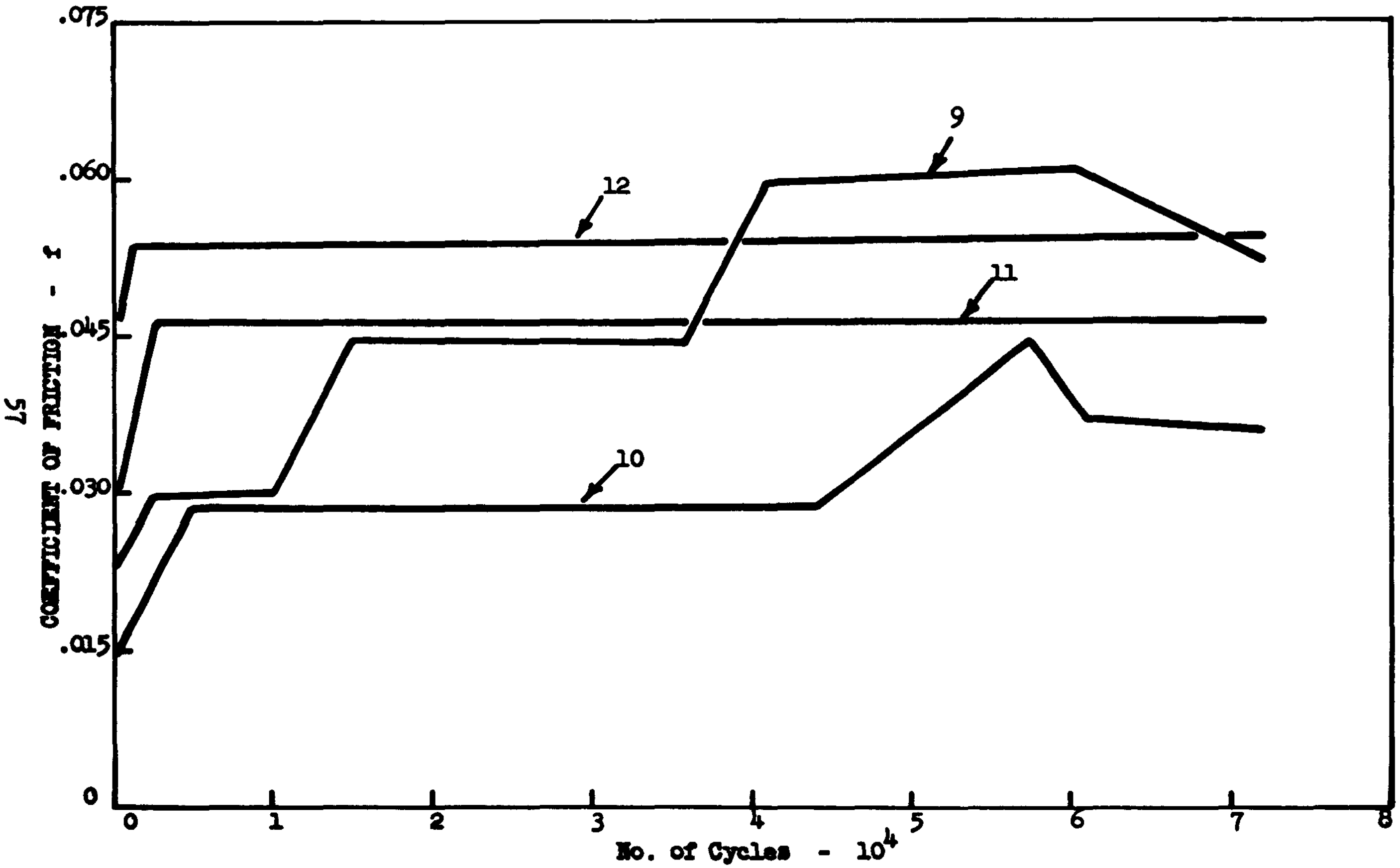


FIGURE 25, NO. OF CYCLES VS FRICTION FOR TESTS 9, 10, 11 & 12 (PHASE II-A-1)

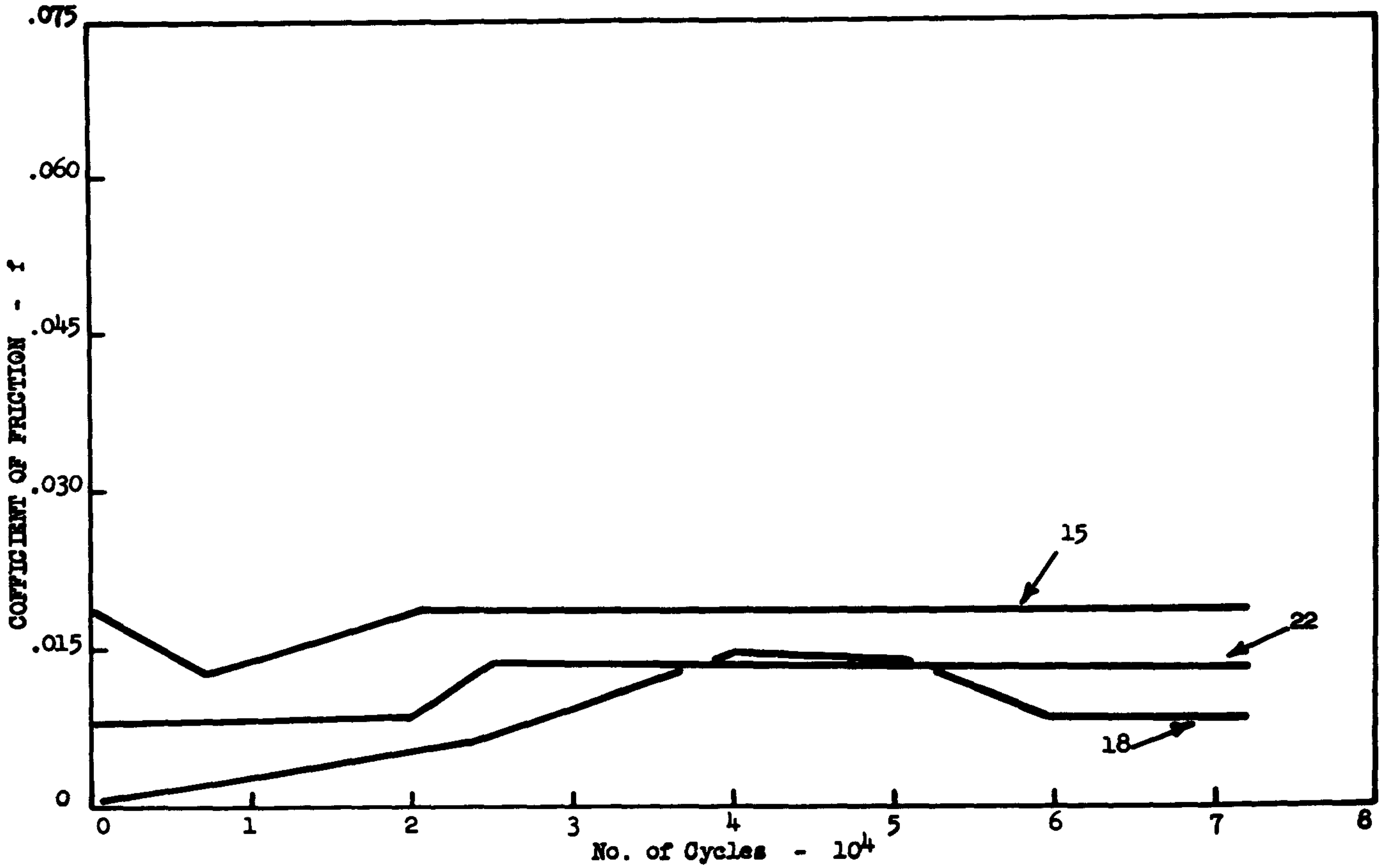


FIGURE 26, NO. OF CYCLES VS FRICTION FOR TESTS 15, 18 & 22 (PHASE II-A-1)

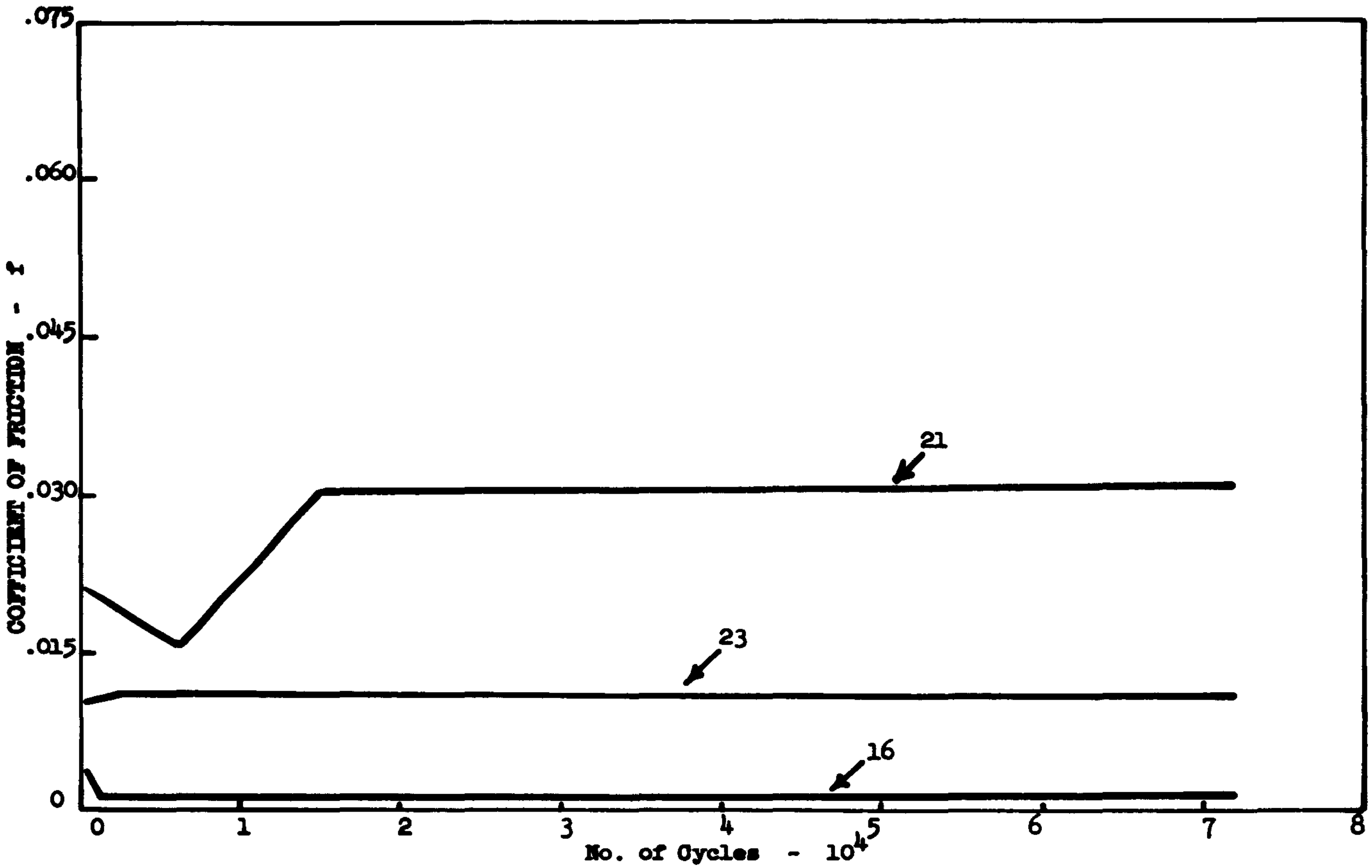


FIGURE 27, NO. OF CYCLES VS FRICTION FOR TESTS 16, 21 & 23 (PHASE II-A-1)

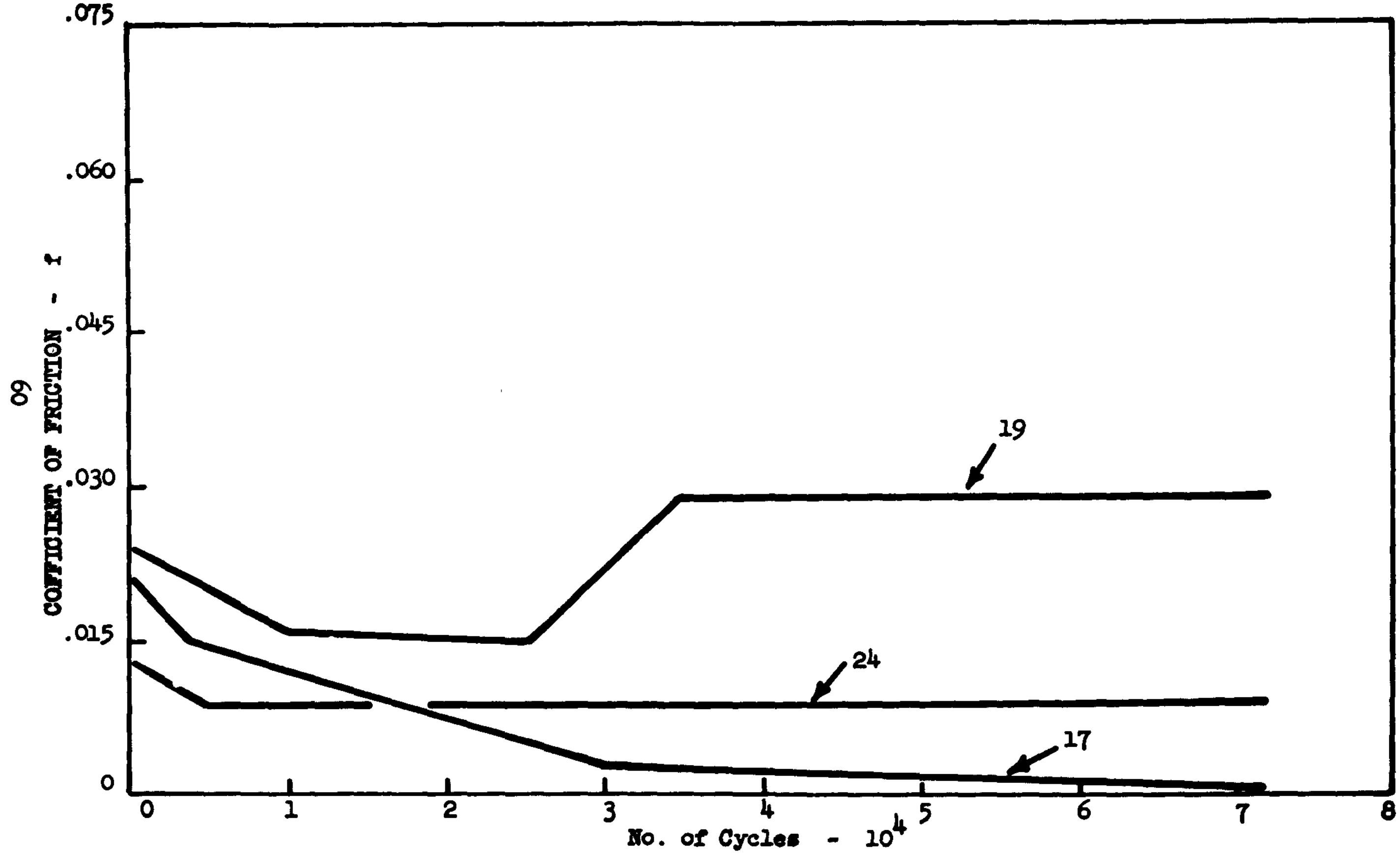


FIGURE 28, NO. OF CYCLES VS FRICTION FOR TESTS 17, 19 & 24 (PHASE II-A-1)

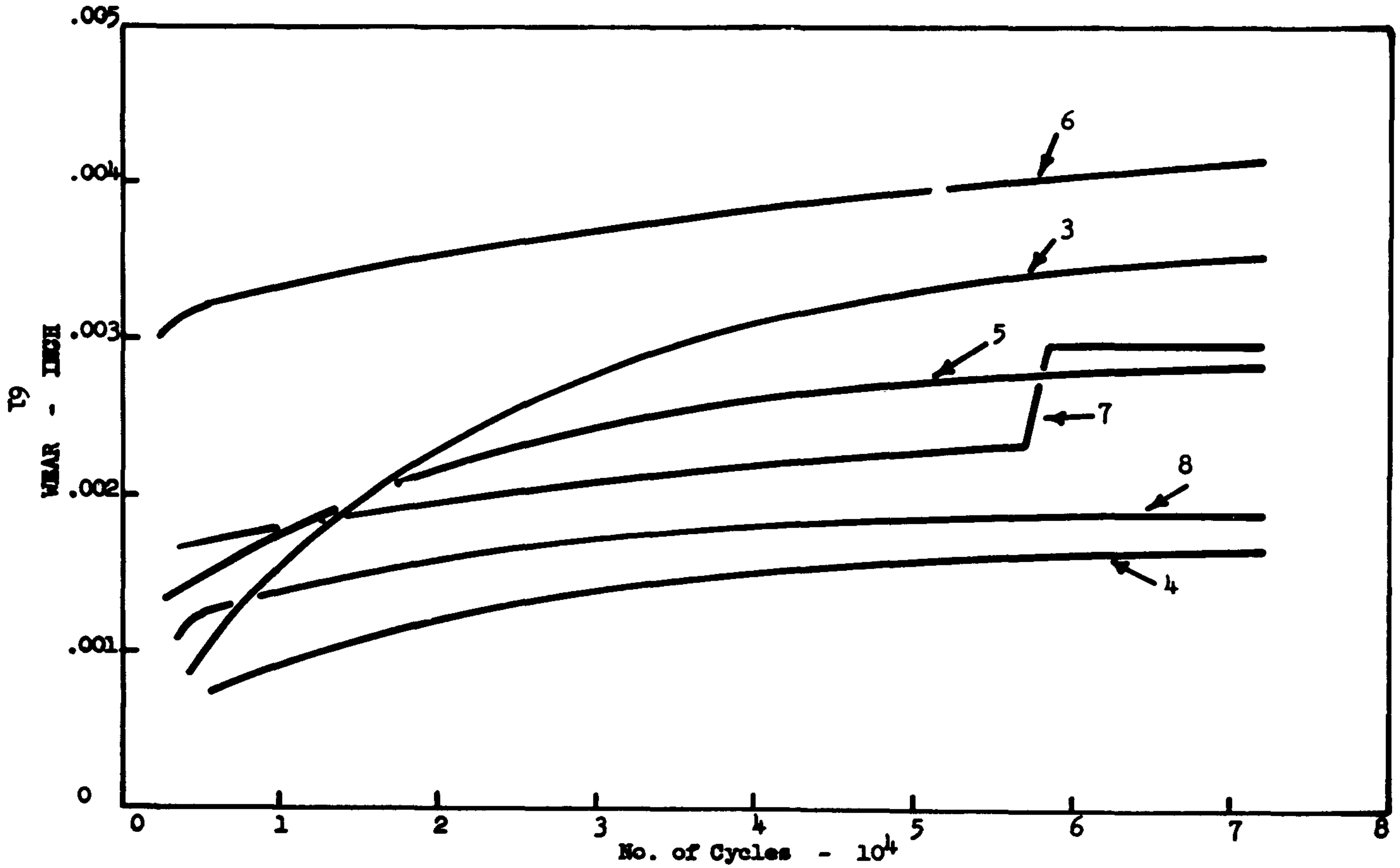


FIGURE 29, NO. OF CYCLES VS WEAR FOR TESTS 3, 4, 5, 6, 7 & 8 (PHASE II-A-2)

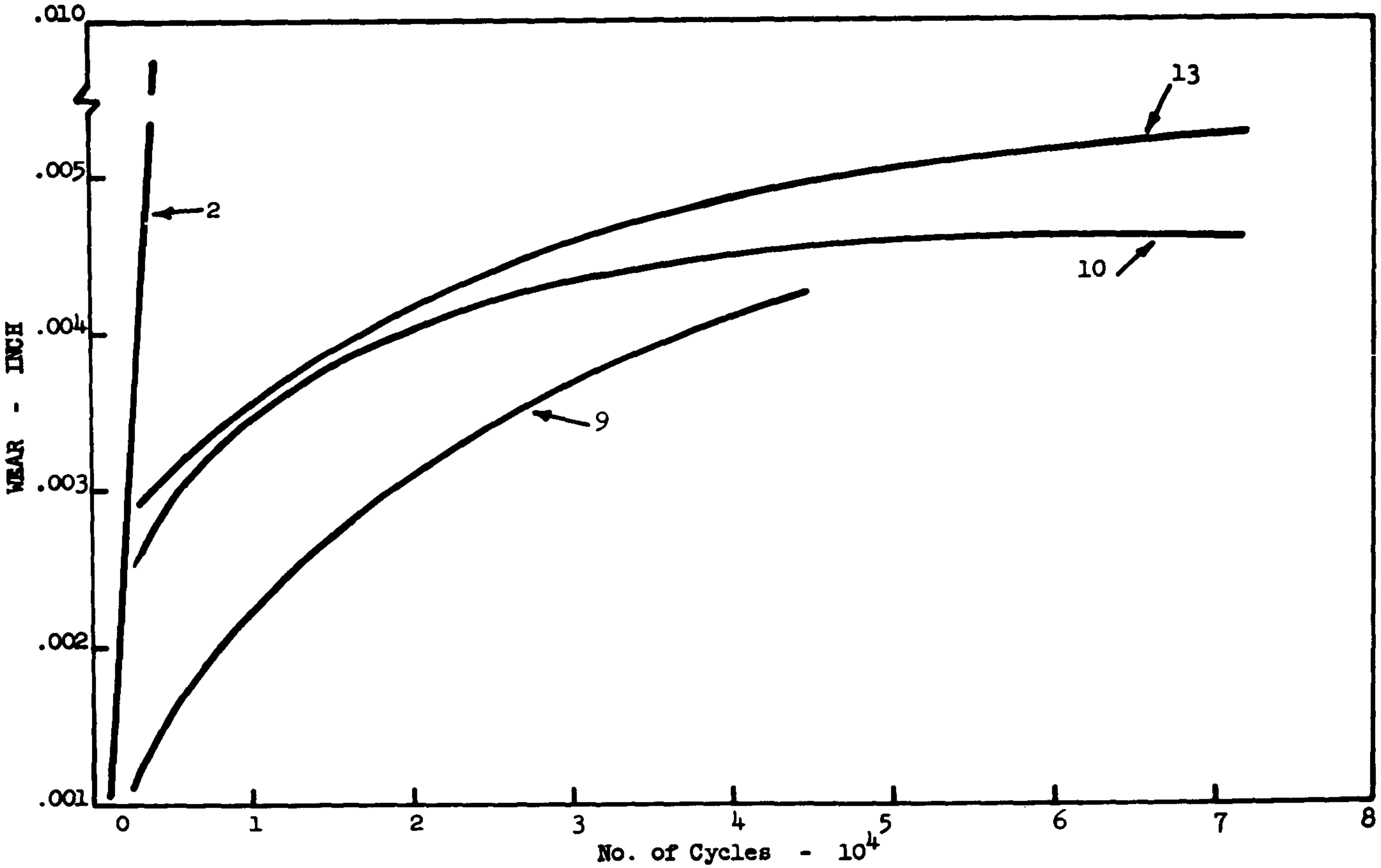


FIGURE 30, NO. OF CYCLES VS WEAR FOR TESTS 2, 9, 10 & 13 (PHASE II-A-2)

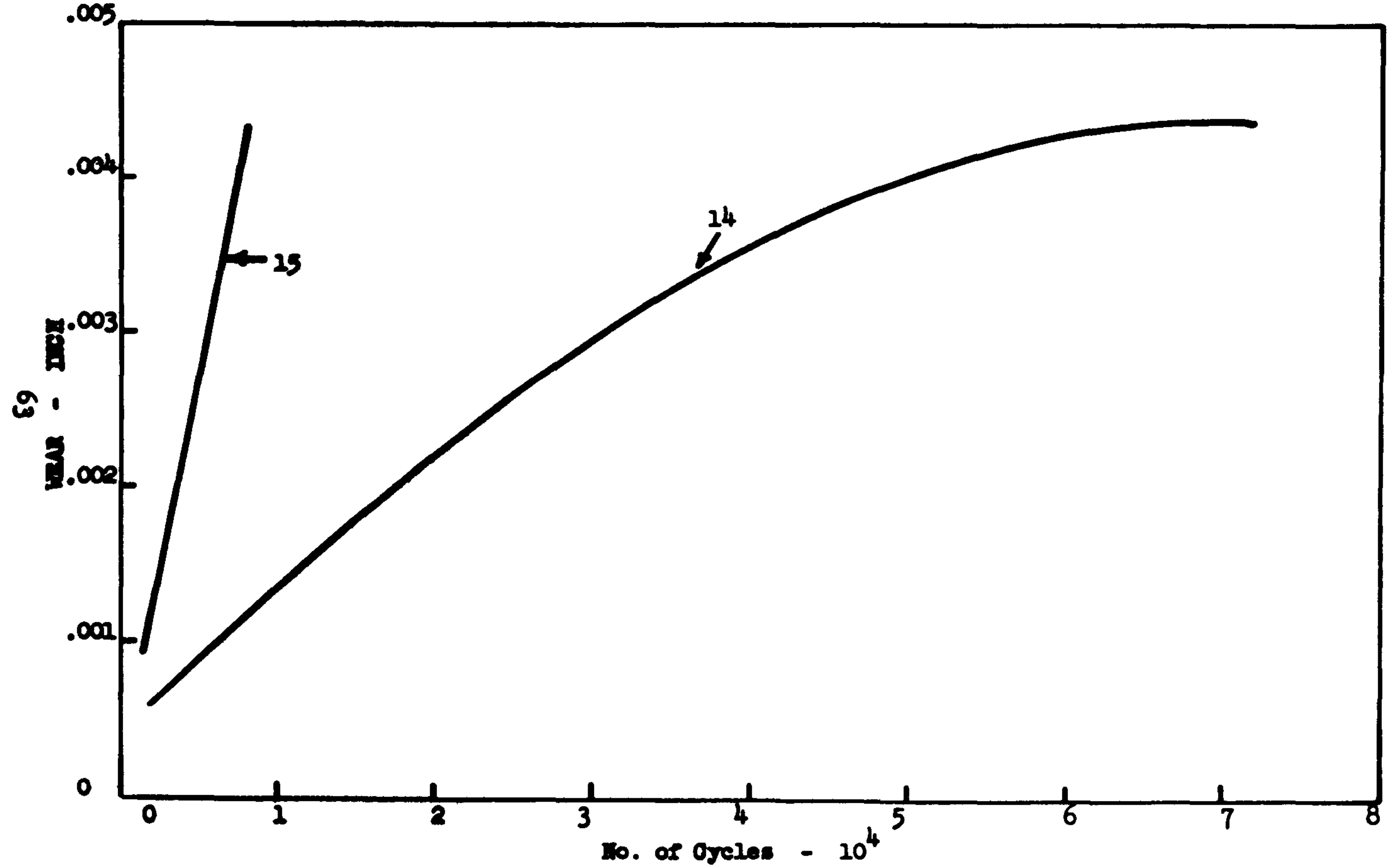


FIGURE 31. NO. OF CYCLES VS WEAR FOR TESTS 14 & 15 (PHASE II-A-2)

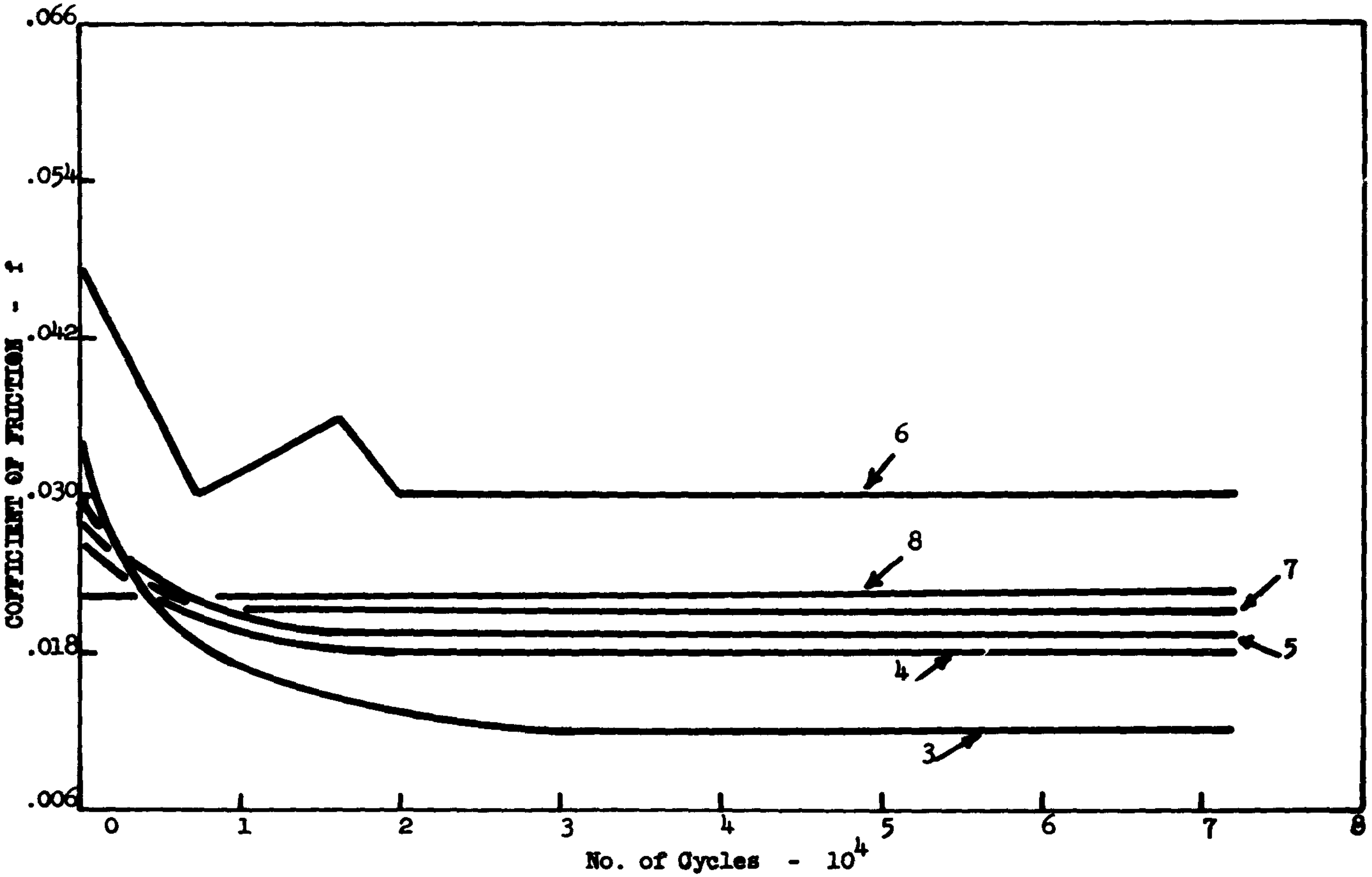


FIGURE 32, NO. OF CYCLES VS FRICTION FOR TESTS 3, 4, 5, 6, 7 & 8 (PHASE II-A-2)

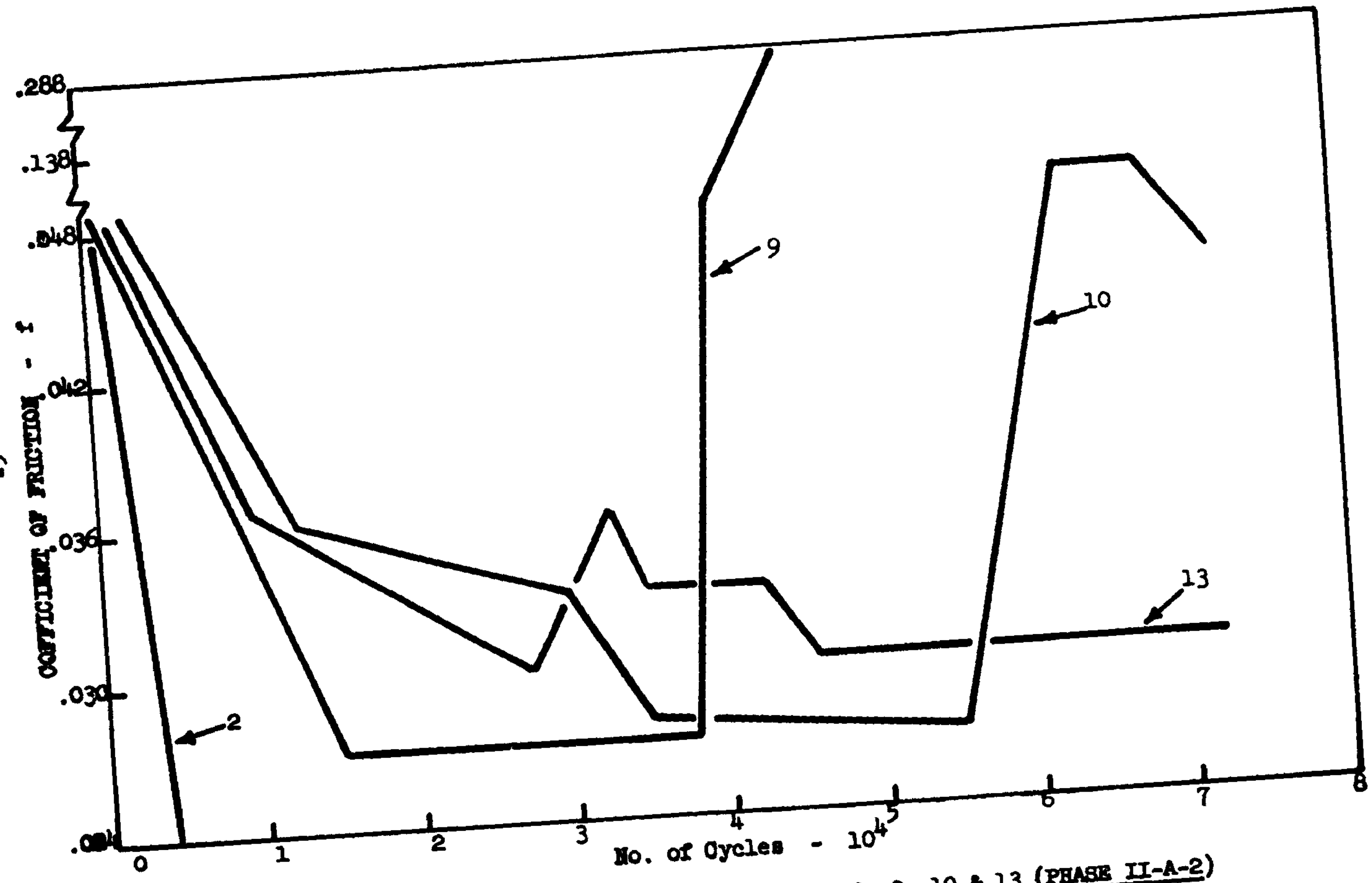


FIGURE 33, NO. OF CYCLES VS FRICTION FOR TESTS 2, 9, 10 & 13 (PHASE II-A-2)

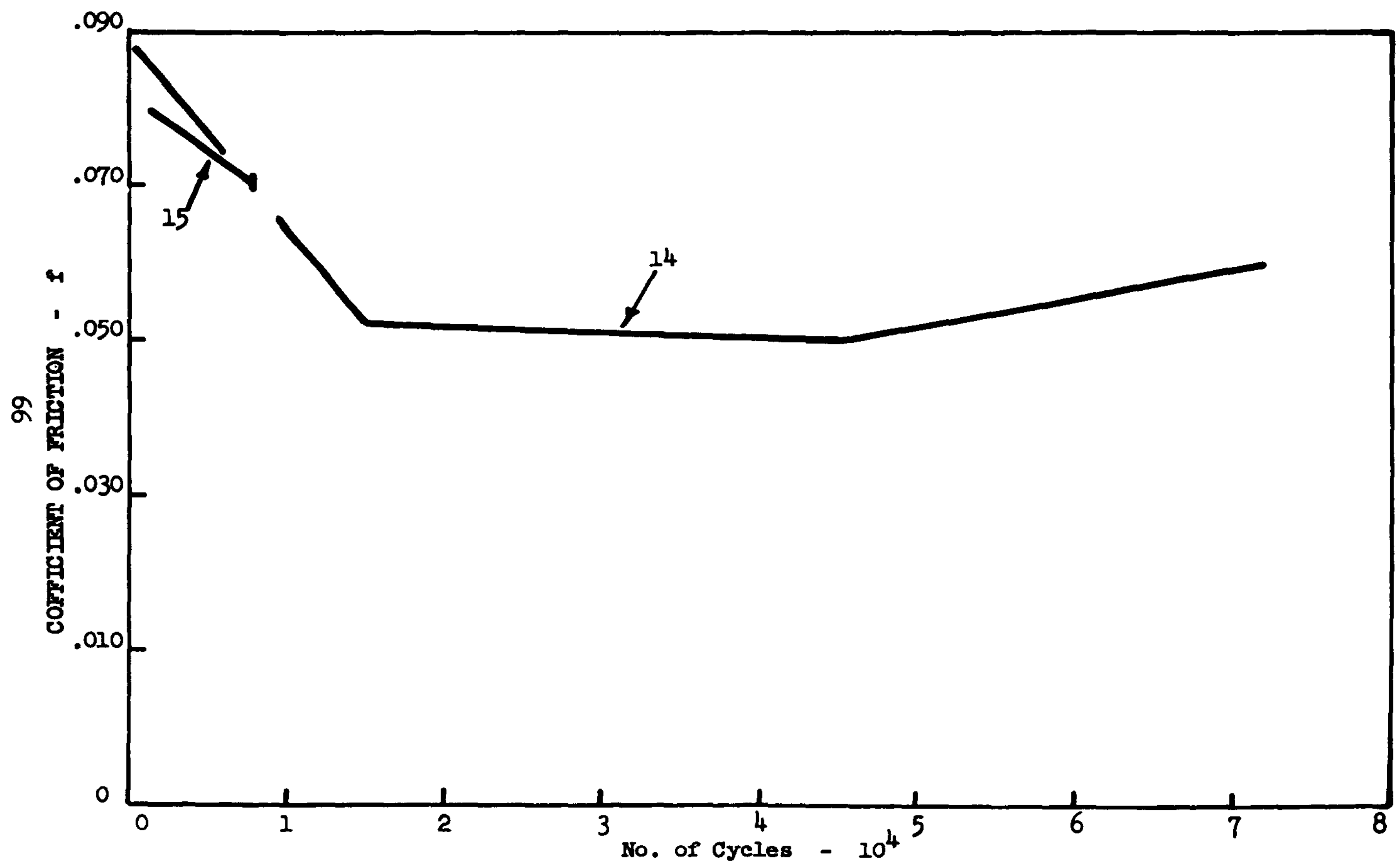


FIGURE 34, NO. OF CYCLES VS FRICTION FOR TESTS 14 & 15 (PHASE II-A-2)

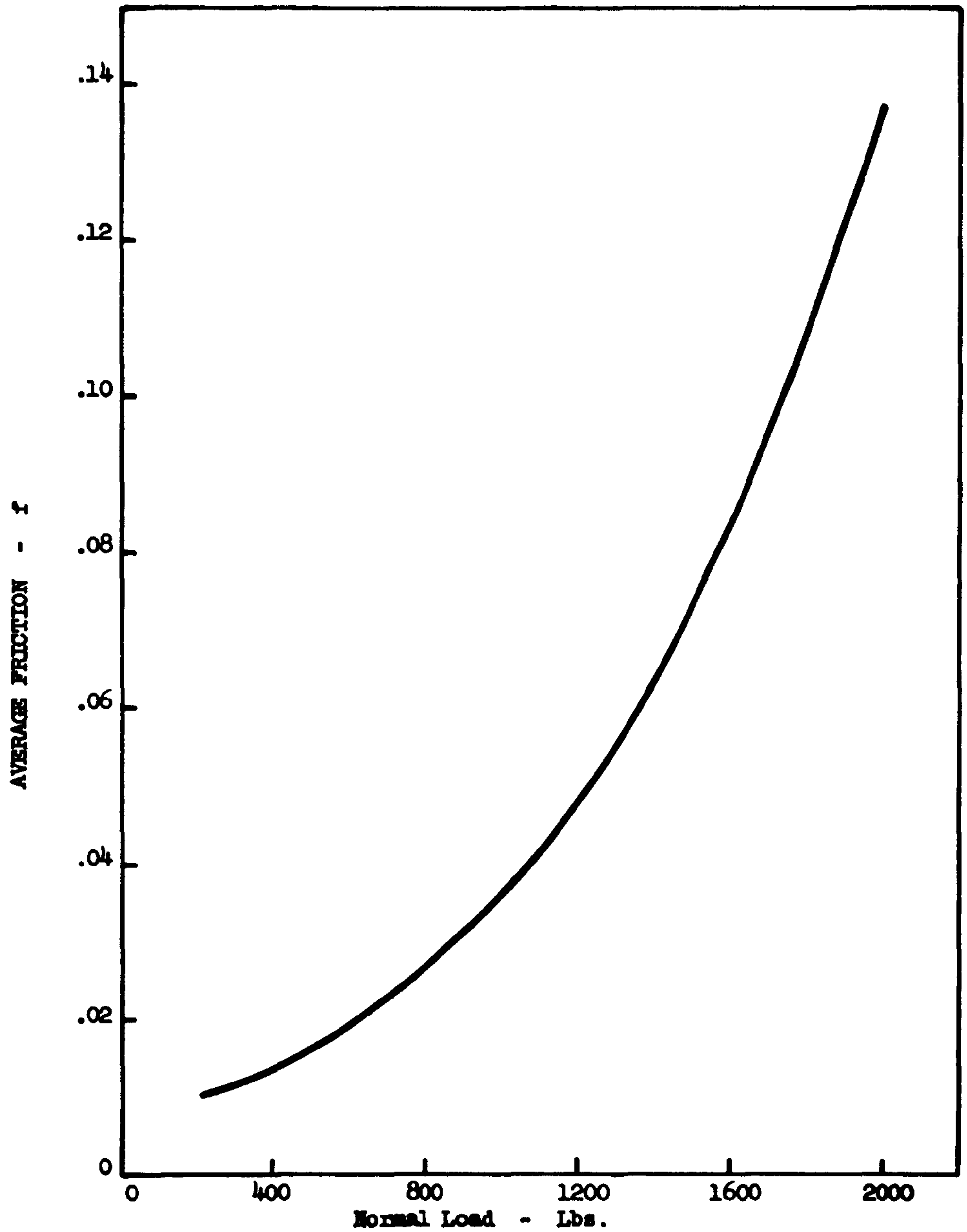
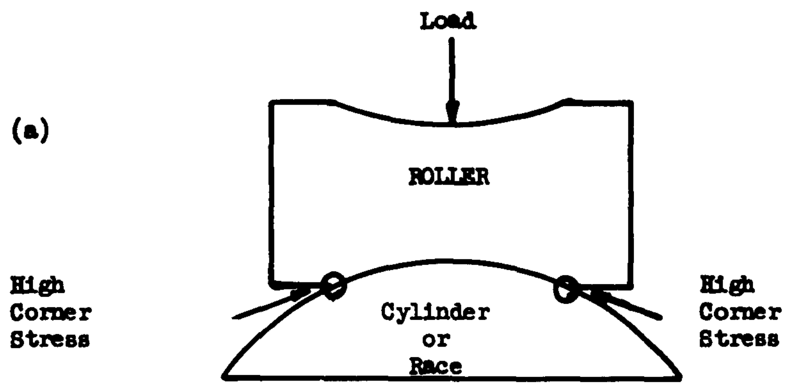
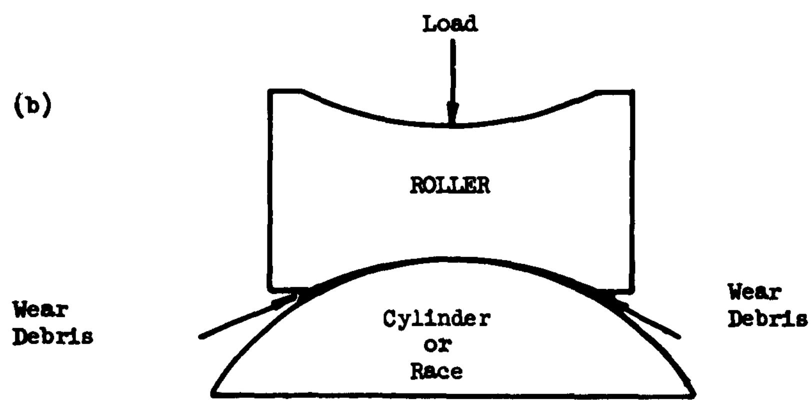


FIGURE 35, LOAD VS AVERAGE FRICTION (PHASE II-A-2)



(a) Excessive corner digging when contact ellipse exceeds available roll surface area. (Very severe when roll material is very hard)



(b) Internal loading and abrasive rubbing when contact ellipse falls short of available roll surface area.

FIGURE 36, ROLL SURFACE CORNER EFFECTS

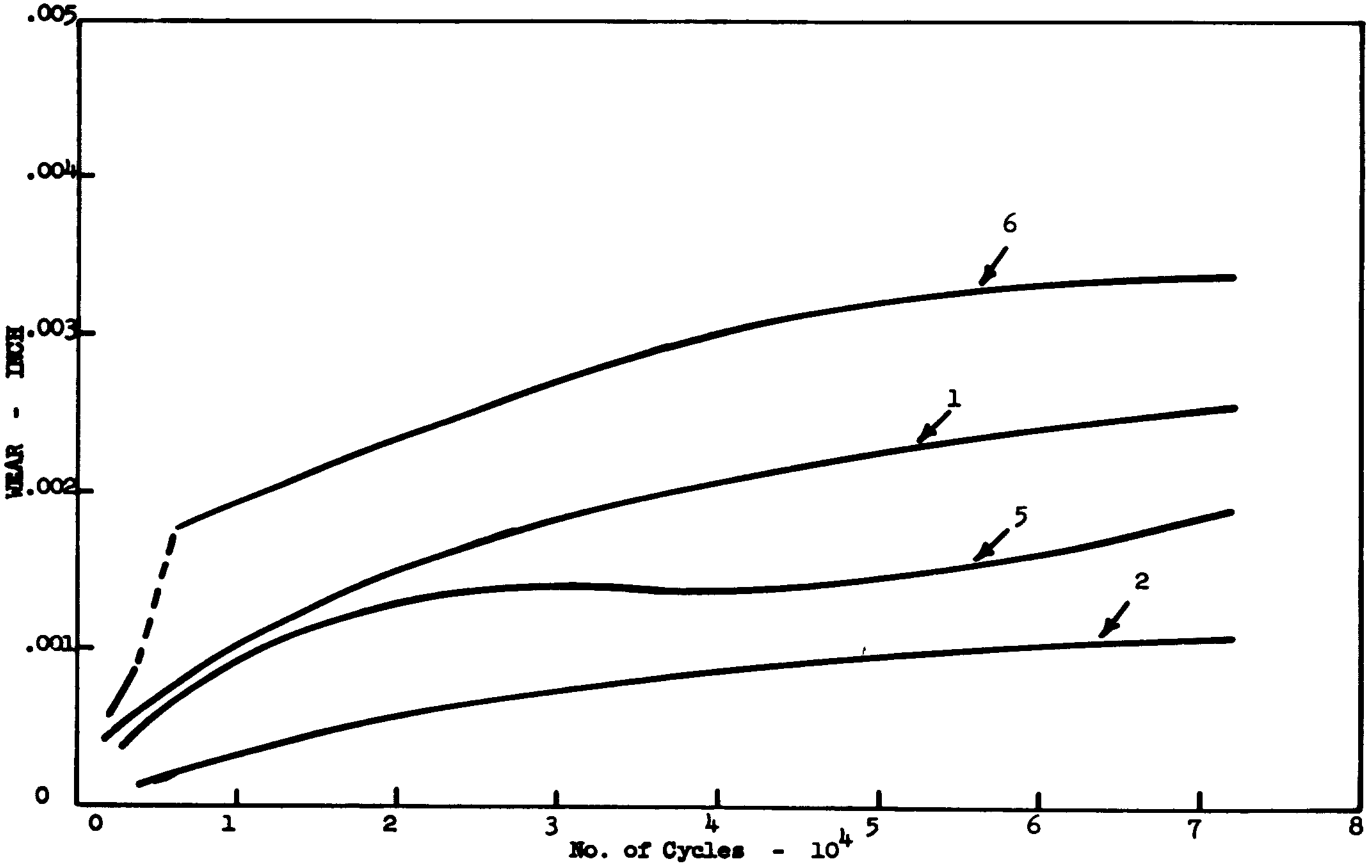


FIGURE 37, NO. OF CYCLES VS WEAR FOR TESTS 1, 2, 5 & 6 (PHASE II-B)

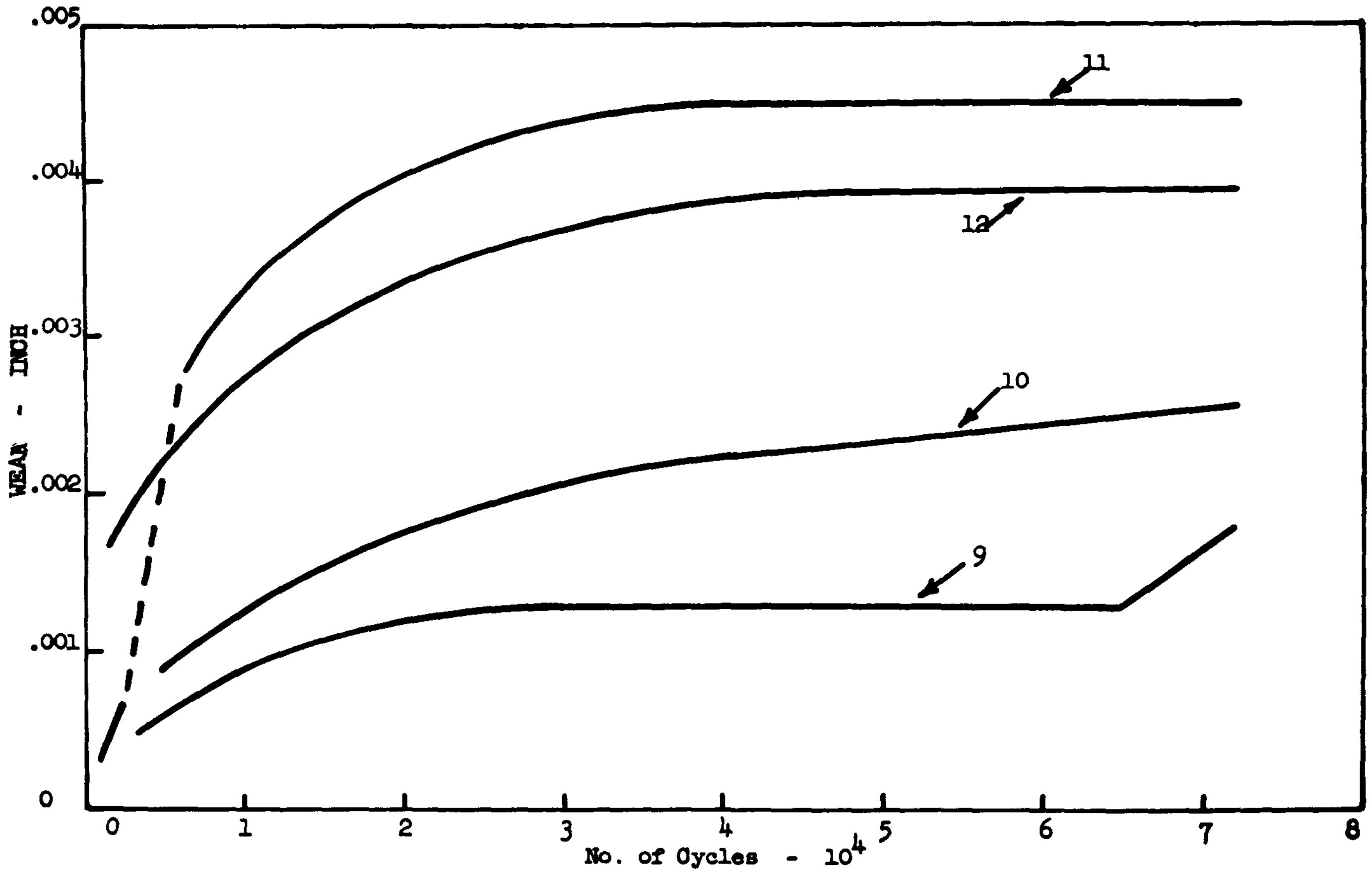


FIGURE 38, NO. OF CYCLES VS WEAR FOR TESTS 9, 10, 11 & 12 (PHASE II-B)

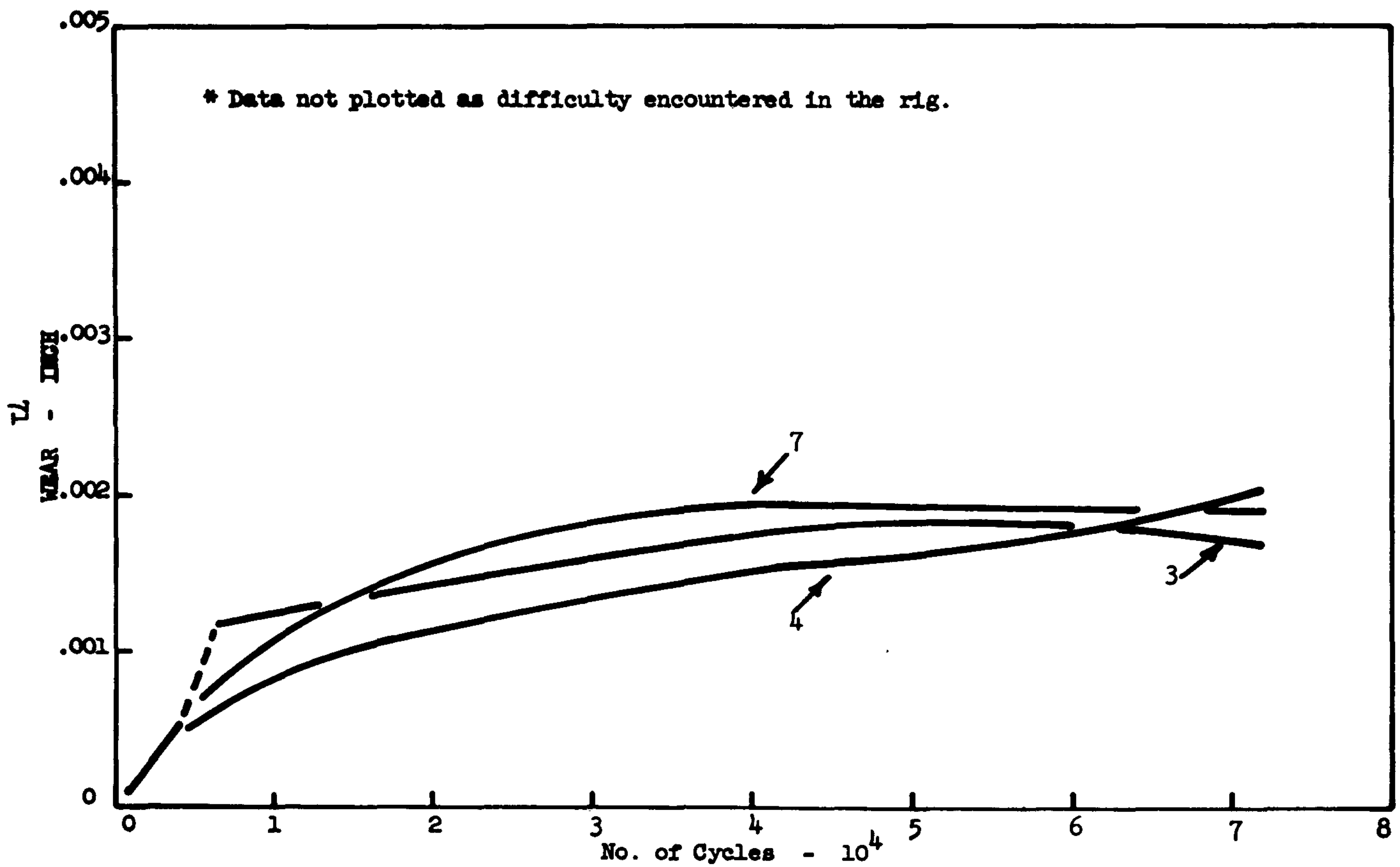


FIGURE 39, NO. OF CYCLES VS WEAR FOR TESTS 3, 4, 7 & 8* (PHASE II-B)

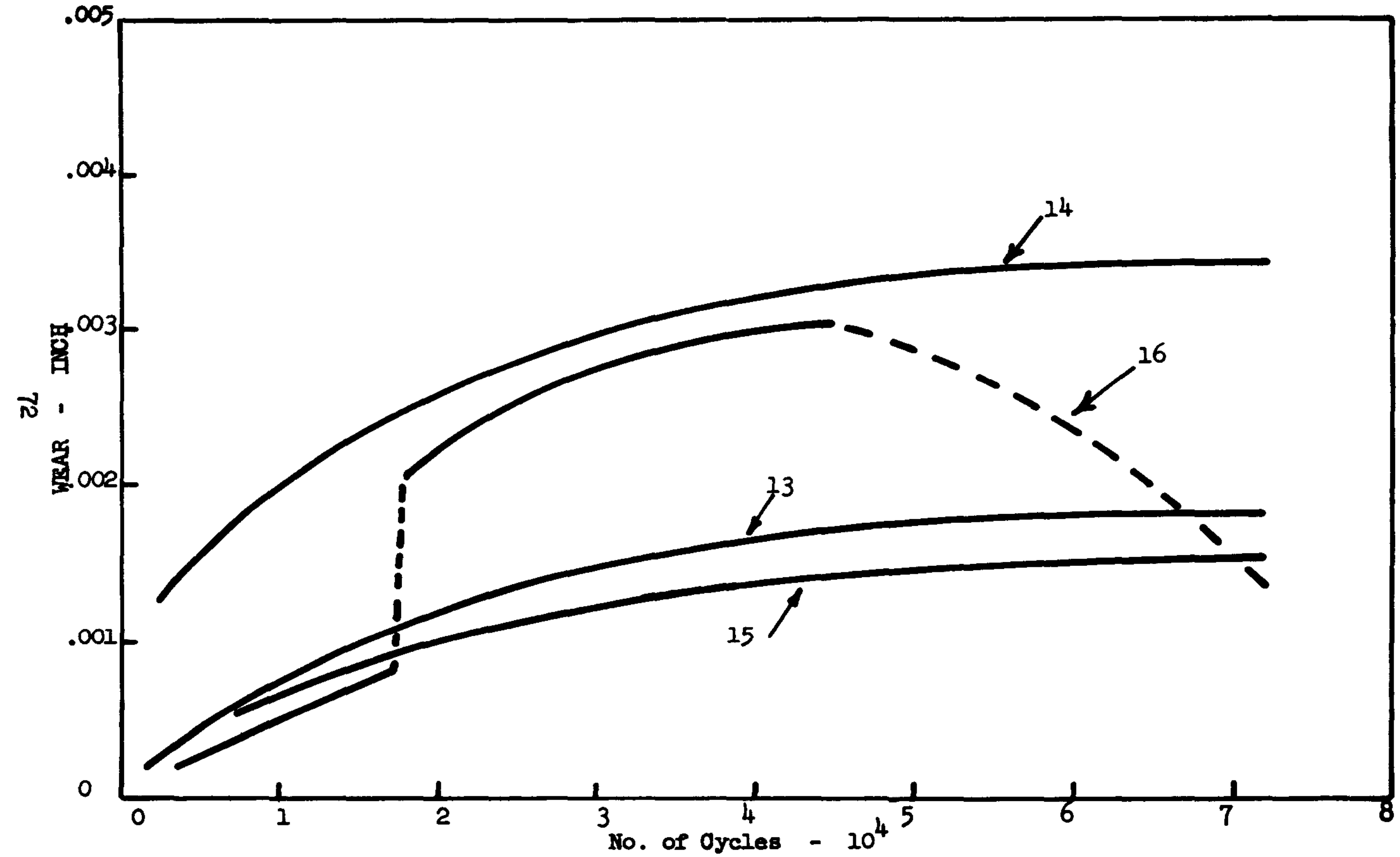


FIGURE 40, NO. OF CYCLES VS WEAR FOR TESTS 13, 14, 15 & 16 (PHASE II-B)

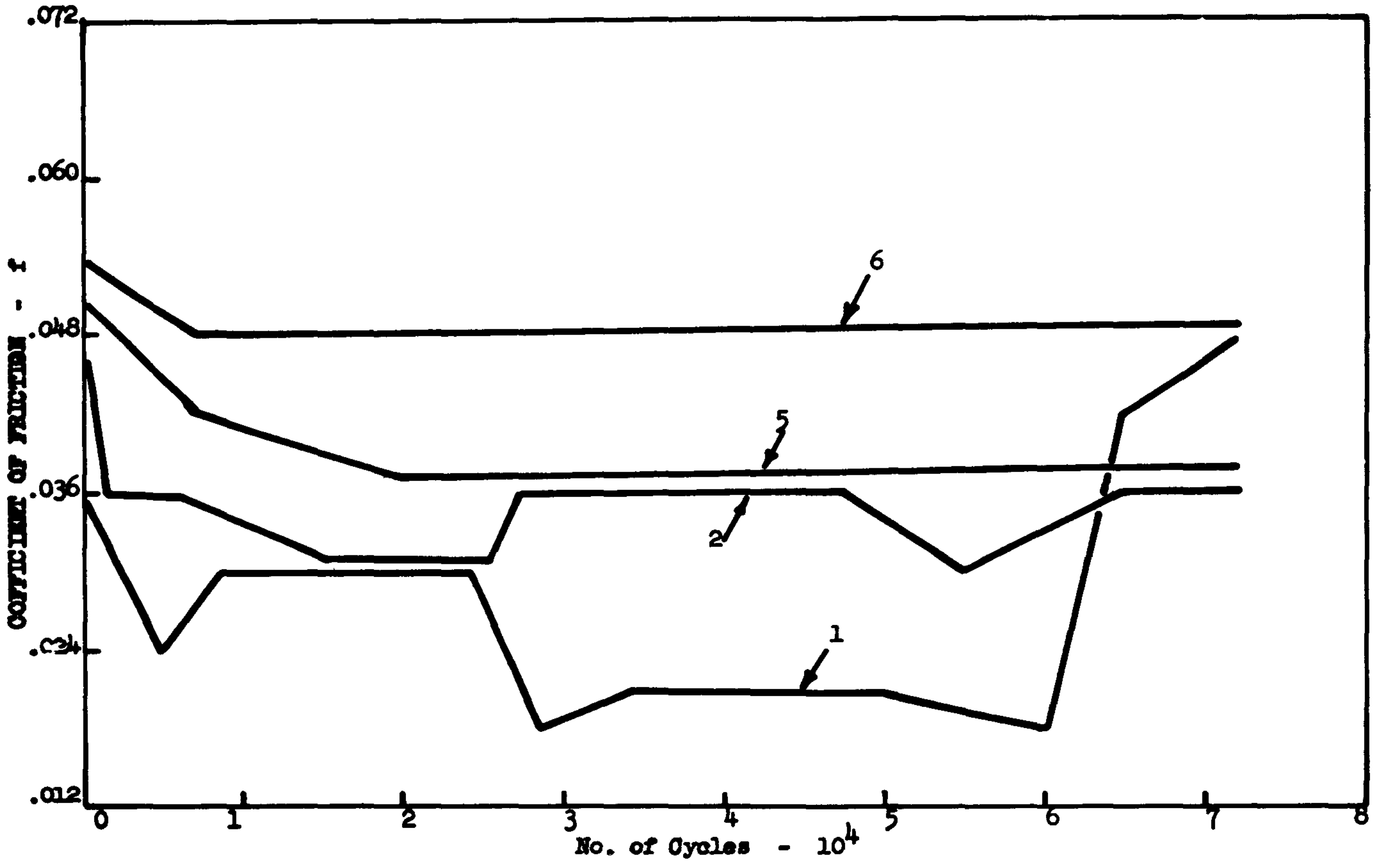


FIGURE 41, NO. OF CYCLES VS FRICTION FOR TESTS 1, 2, 5 & 6 (PHASE II-B)

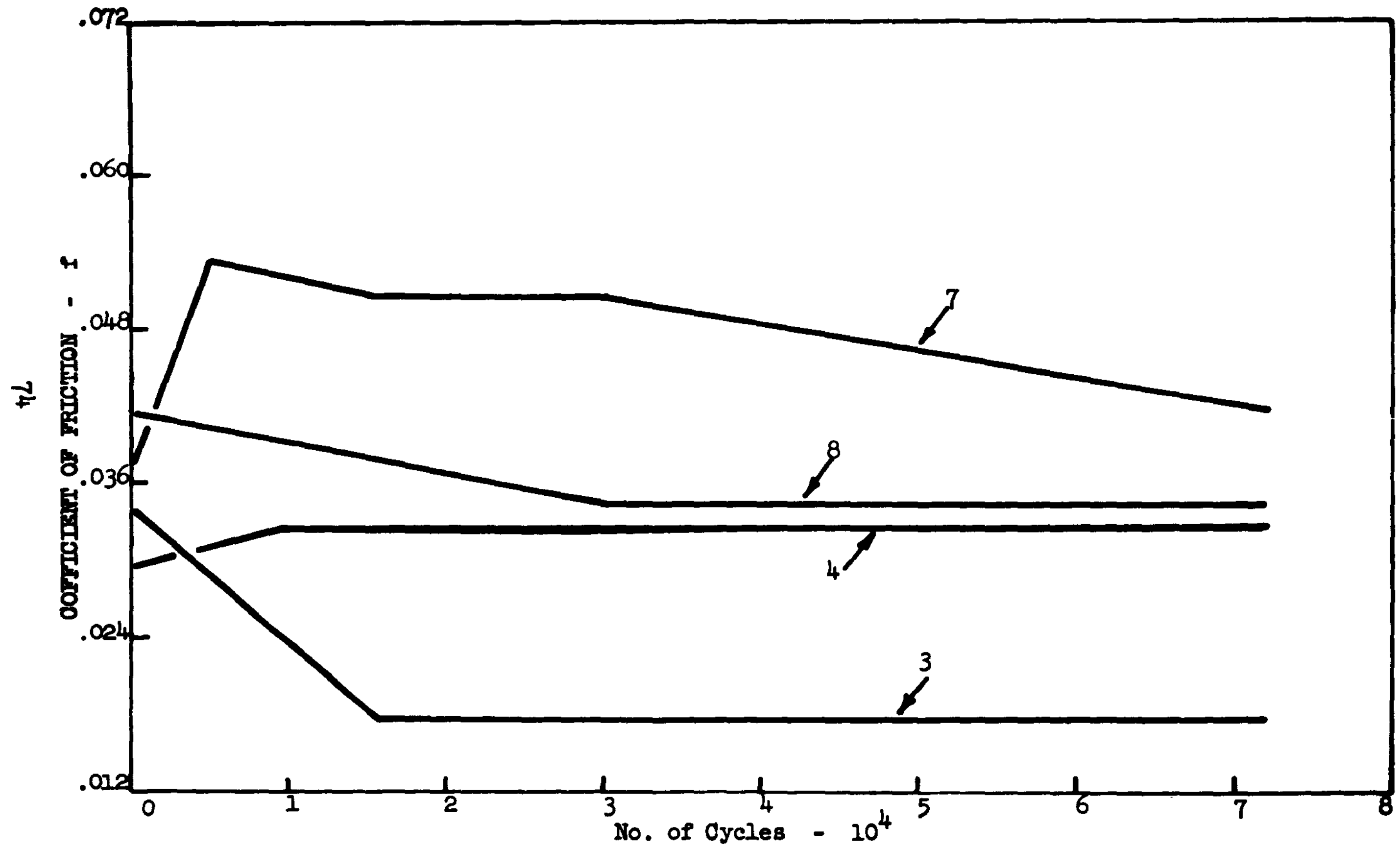


FIGURE 42, NO. OF CYCLES VS FRICTION FOR TEST 3, 4, 7 & 8 (PHASE II-B)

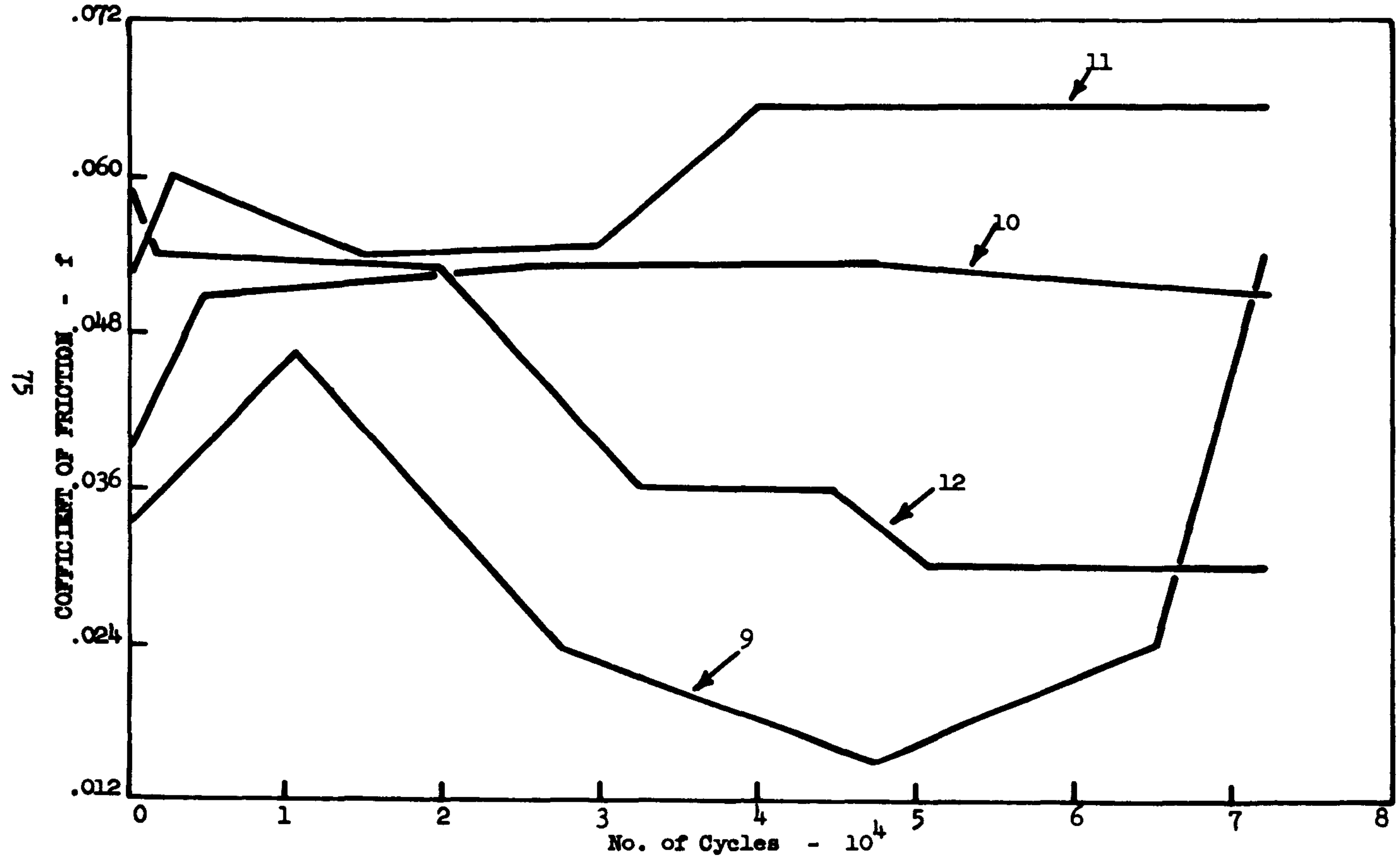


FIGURE 43, NO. OF CYCLES VS FRICTION FOR TESTS 9, 10, 11 & 12 (PHASE II-B)

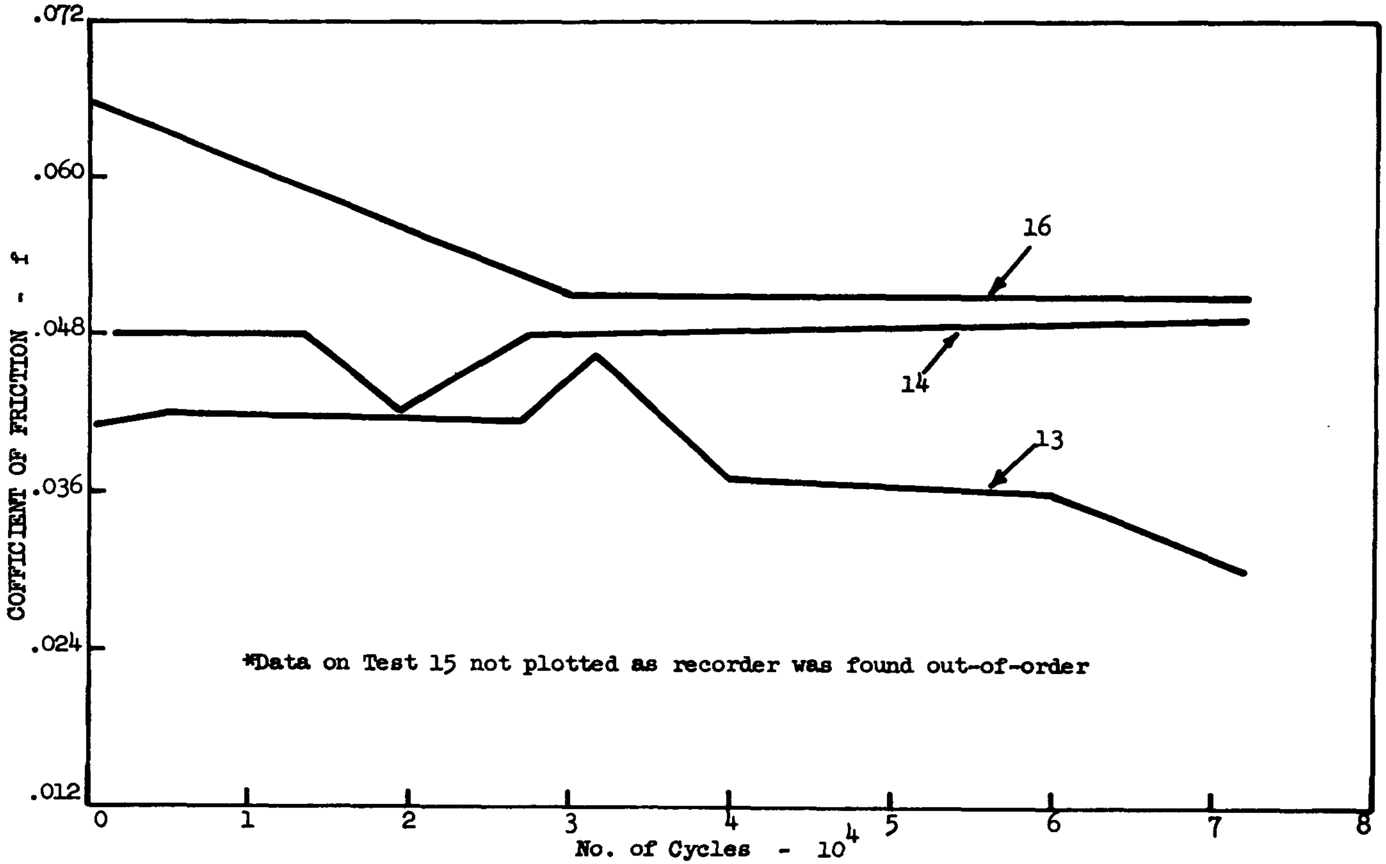


FIGURE 44, NO. OF CYCLES VS FRICTION FOR TESTS 13, 14, 15* & 16 (PHASE II-B)

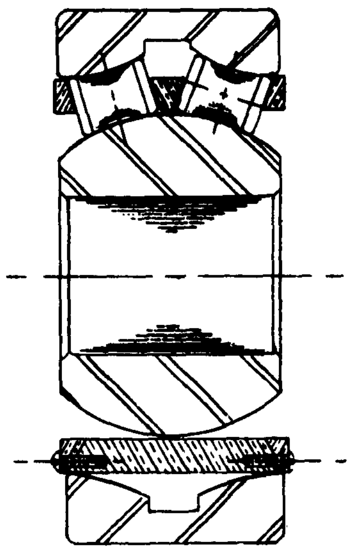


FIGURE 45, BR-16 DOUBLE ROW BEARING

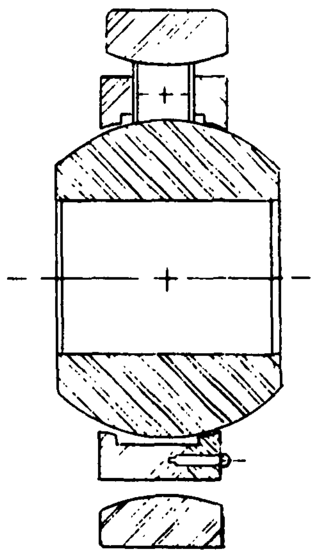
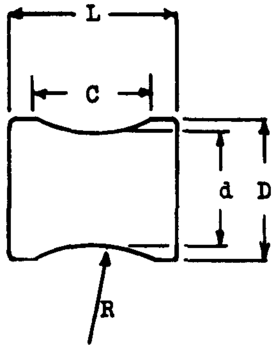


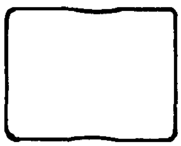
FIGURE 46, BR-16 SINGLE ROW BEARING



Test No.

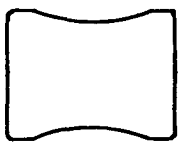
D = .4331"
d = .3978"
R =1.0050"
C = .375 "
L = .5354"

1, 2, 3, 4,
5, 6, 7



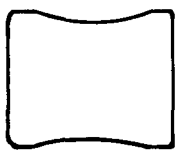
D = .4068"
d = .3978"
R =1.0050"
C = .1900"
L = .5354"

8, 9



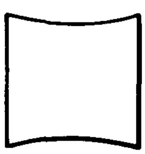
D = .4308"
d = .3978"
R =1.0640"
C = .3750"
L = .5354"

10, 11, 12



D = .4205"
d = .3970"
R =1.4970"
C = .3750"
L = .5354"

13, 14



D = .4205"
d = .3970"
R =1.4970"
C = .3750"
L = .4150"

15, 16, 17, 18

FIGURE 47, DIFFERENT ROLL DESIGNS

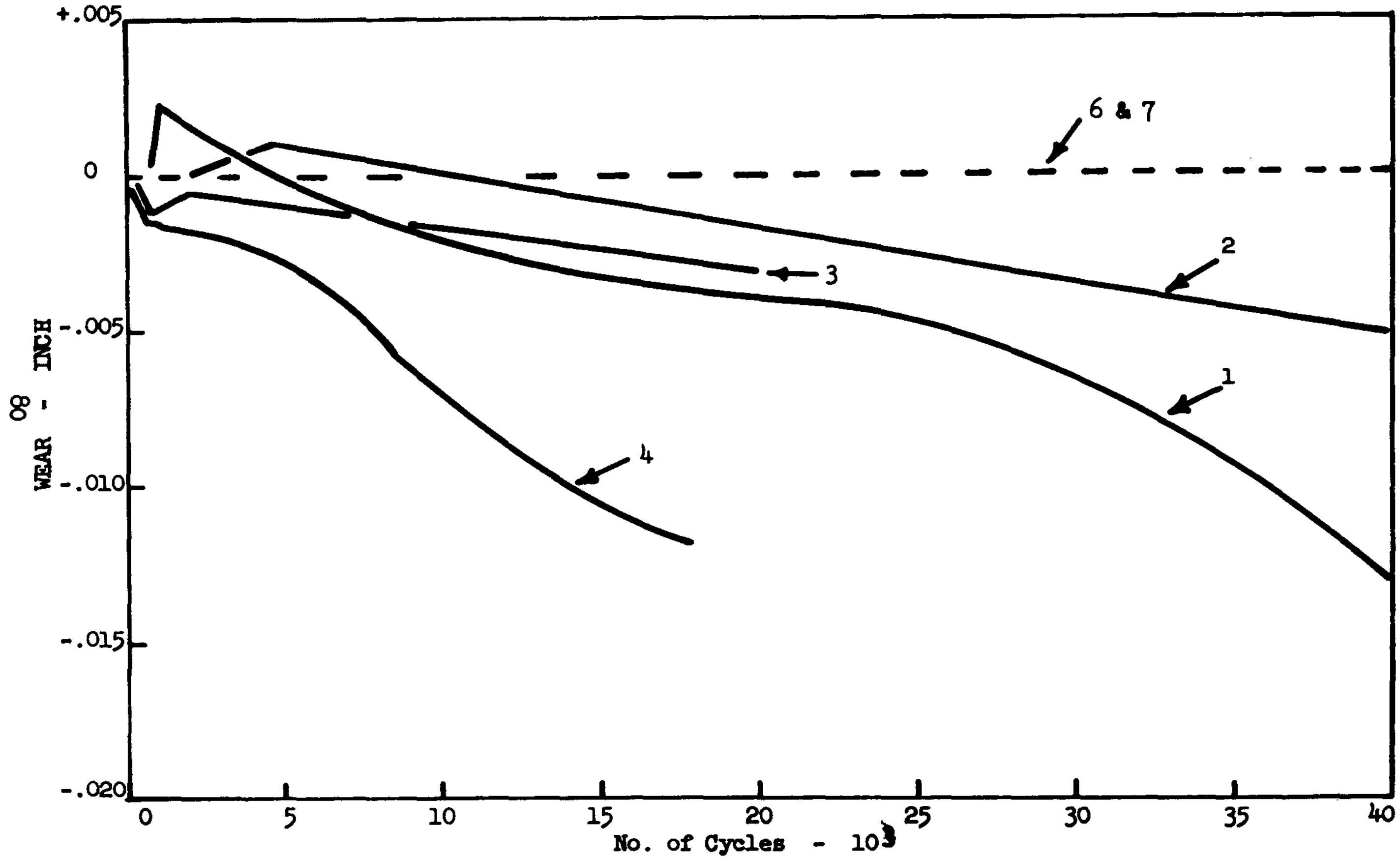


FIGURE 48, NO. OF CYCLES VS WEAR FOR TESTS 1, 2, 3, 4, 6 & 7 (PHASE III)

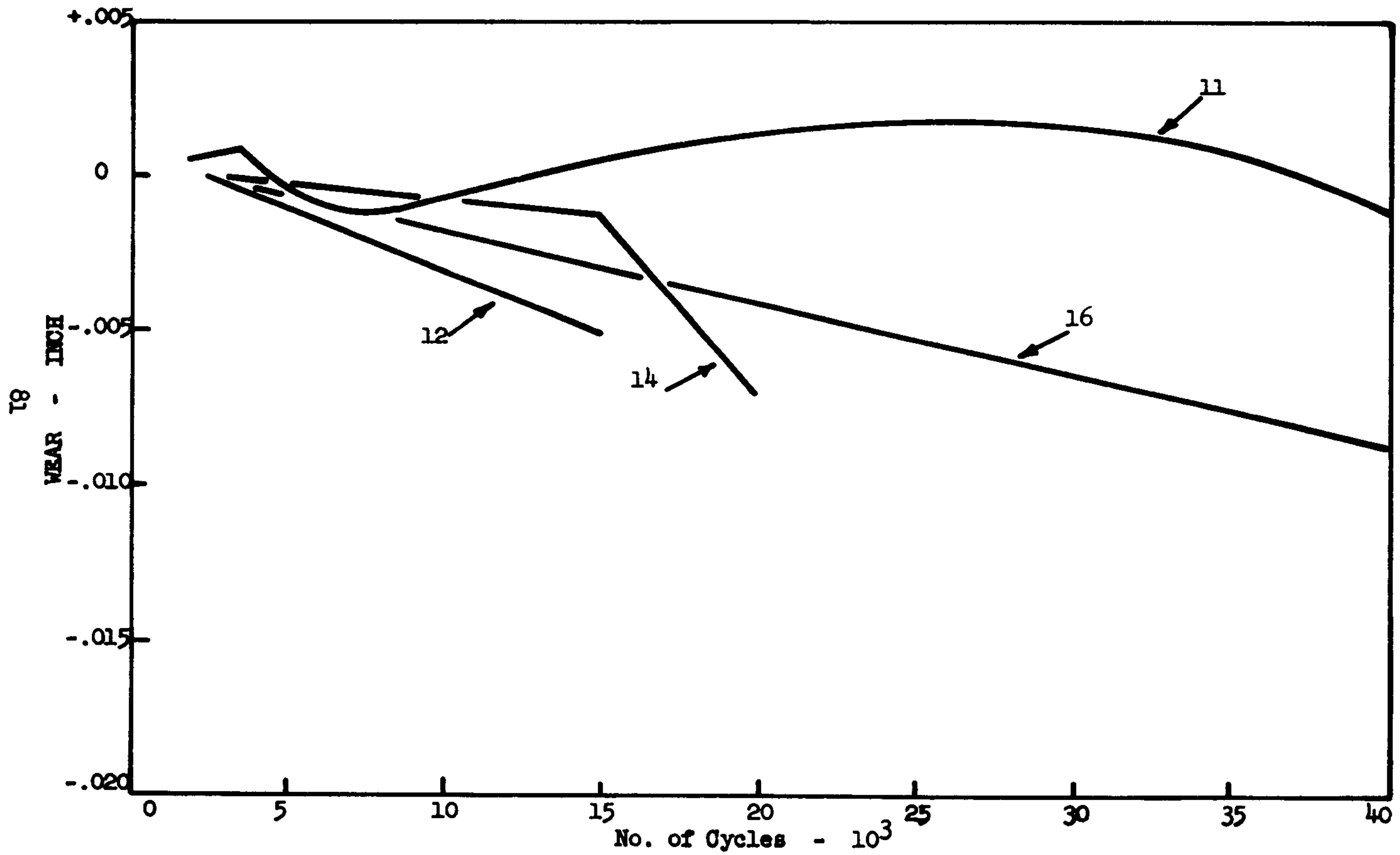


FIGURE 49, NO. OF CYCLES VS WEAR FOR TESTS 11, 12, 14 & 16 (PHASE III)

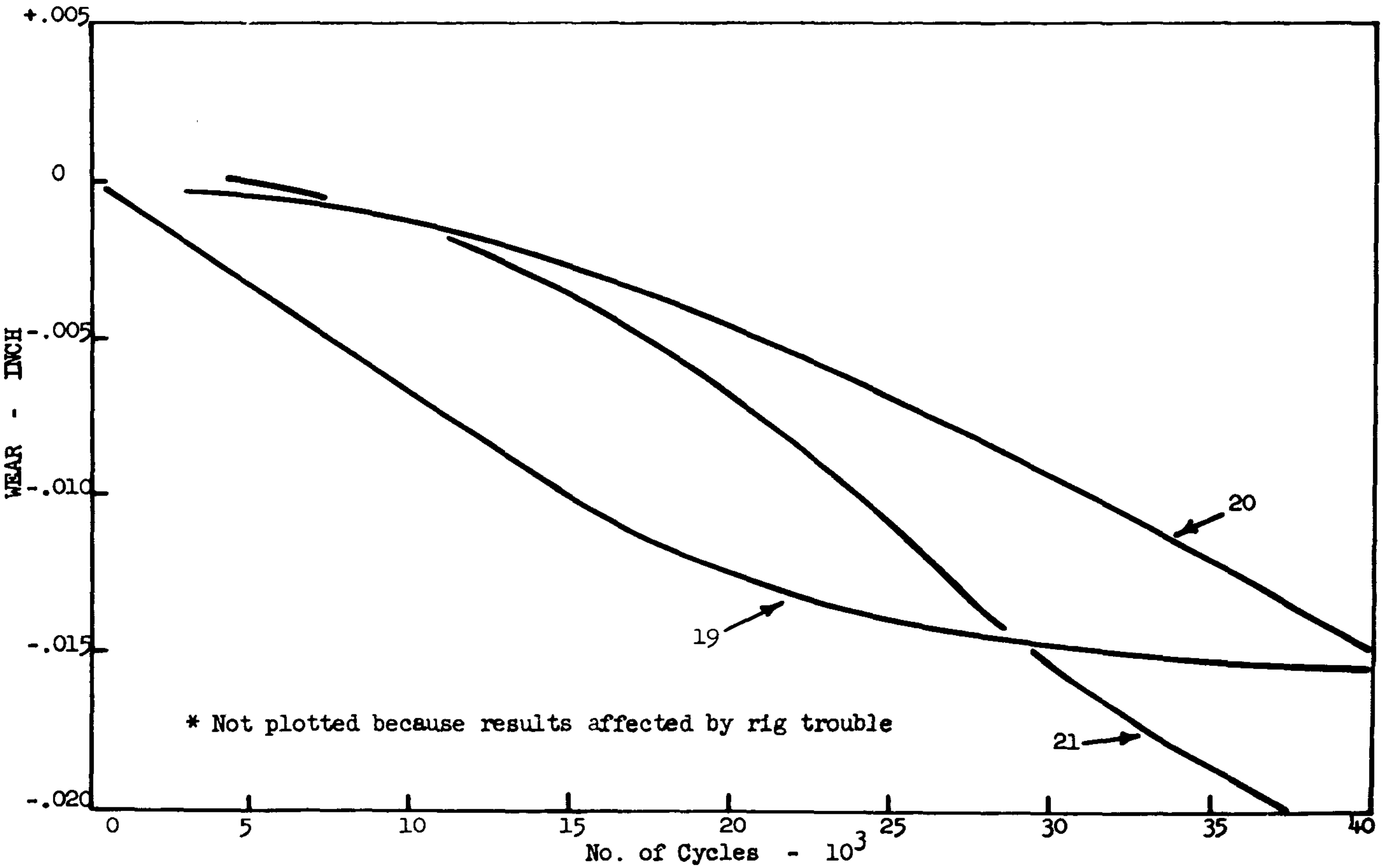


FIGURE 50, NO. OF CYCLES VS WEAR FOR TESTS 17* 18*, 19, 20 & 21 (PHASE III)

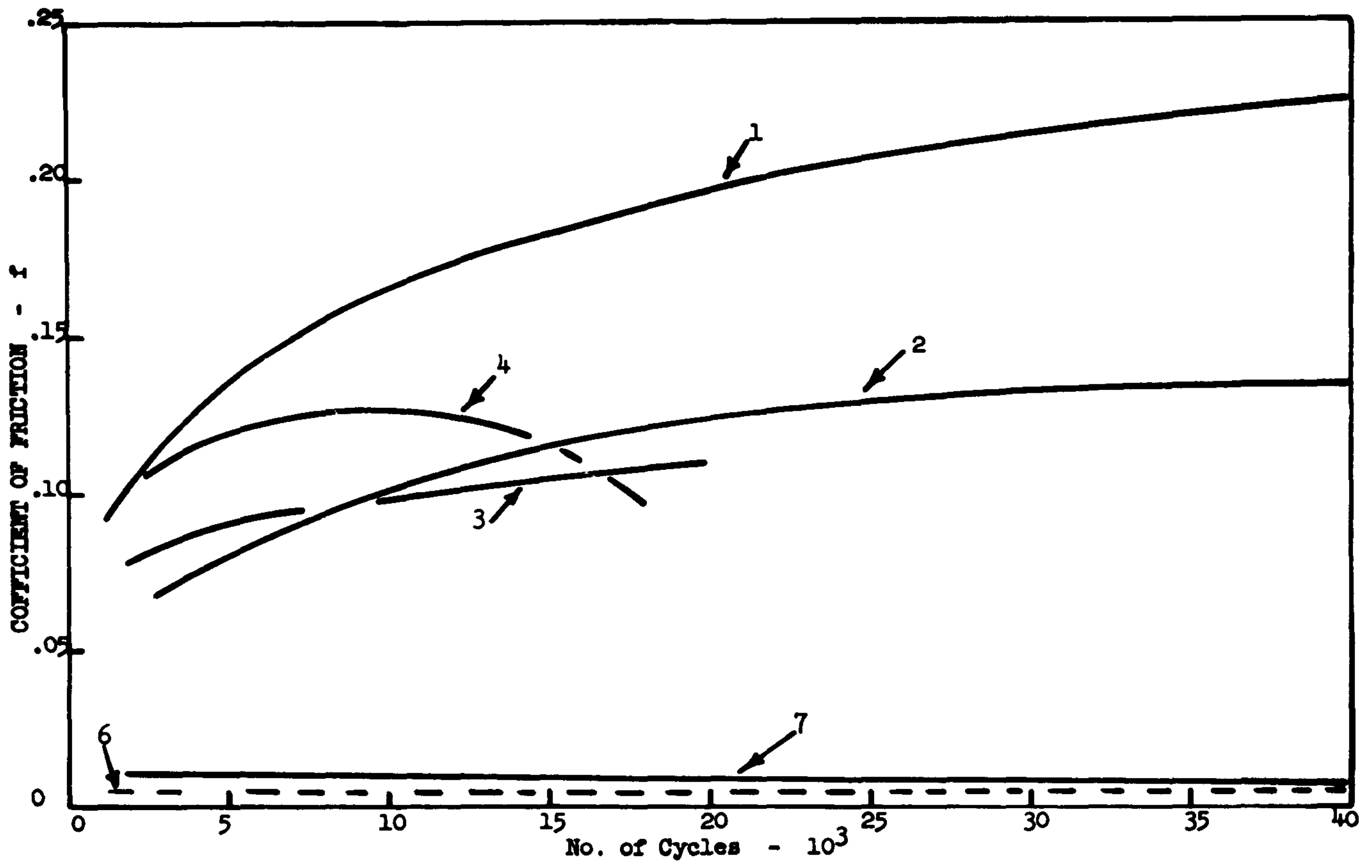


FIGURE 51, NO. OF CYCLES VS FRICTION FOR TESTS 1, 2, 3, 4, 6 & 7 (PHASE III)

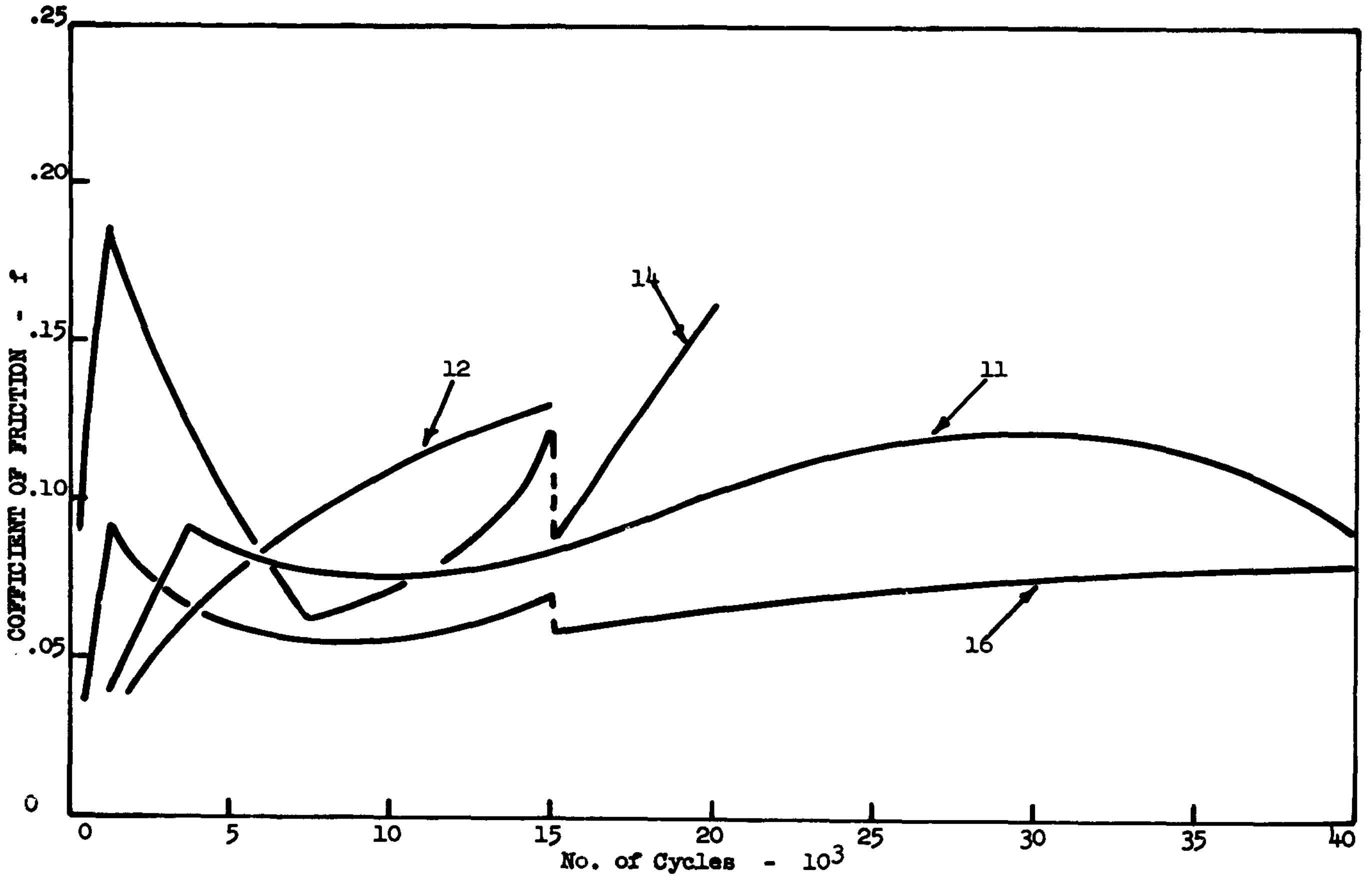


FIGURE 52, NO. OF CYCLES VS FRICTION FOR TESTS 11, 12, 14 & 16 (PHASE III)

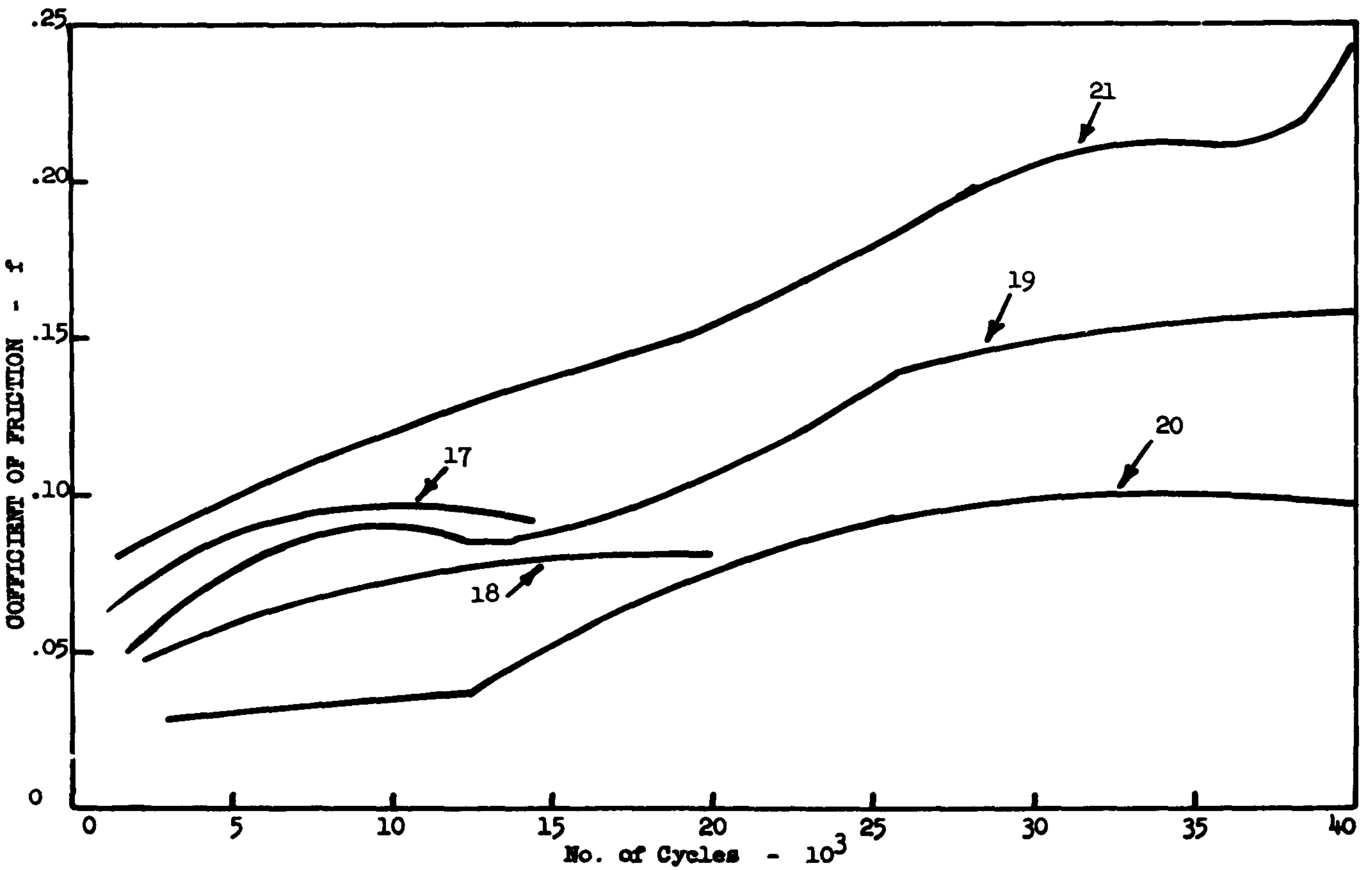


FIGURE 53, NO. OF CYCLES VS FRICTION FOR TESTS 17, 18, 19, 20 & 21 (PHASE III)

Aeronautical Systems Division, Dir/Aero-mechanics Flight Dynamics Lab, Wright-Patterson AFB, Ohio.
Rpt Nr ASD-TDR-62-900. DESIGN CRITERIA FOR ROLLING ELEMENT AIR FRAME BEARINGS FOR HIGH TEMPERATURE AND HIGH ALTITUDE USE. Final Report, Apr 63, 85p. incl illus, tables, 5 refs.

Unclassified Report

Aircraft Control Bearings were experimentally studied to evaluate bearing design, materials, and lubricants for operation at 1200°F and 250,000 ft. altitude.

Four roll designs, twelve material combinations and five dry lubricants were investigated. The best design and two best

(over)

1. Roller Bearings
2. High Temperature Research
3. High Altitude
4. Design
I. AFSC Project 1315 Task 131501
II. Contract AF33(616)-6650
III. Marlin-Rockwell Corp Jamestown, N. Y.
IV. Harold E. Munson, Jamshed B. Havewala John H. Johnson
V. MRC Report 1299
VI. Avail. for OTS
VII. In ASTIA Collection

material combinations (6B (Cobalt Alloy) vs 6B and CA-3 (Tungsten Carbide Cermet) vs 6B) were subjected to stresses up to 325,000 psi at the temperature and altitude.

One inch diameter bore, self aligning, double row roller bearings fabricated from 6B vs 6B carried loads to 5000 lbs. for 40,000 cycles at 1200°F and 250,000 ft. altitude when lubricated with DF-700 dry film. Friction coefficients with the lubricants ranged from .08 to .25. Successful bearing operation requires considerable deviation from design criteria for fluid lubricated bearings.

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